The
ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

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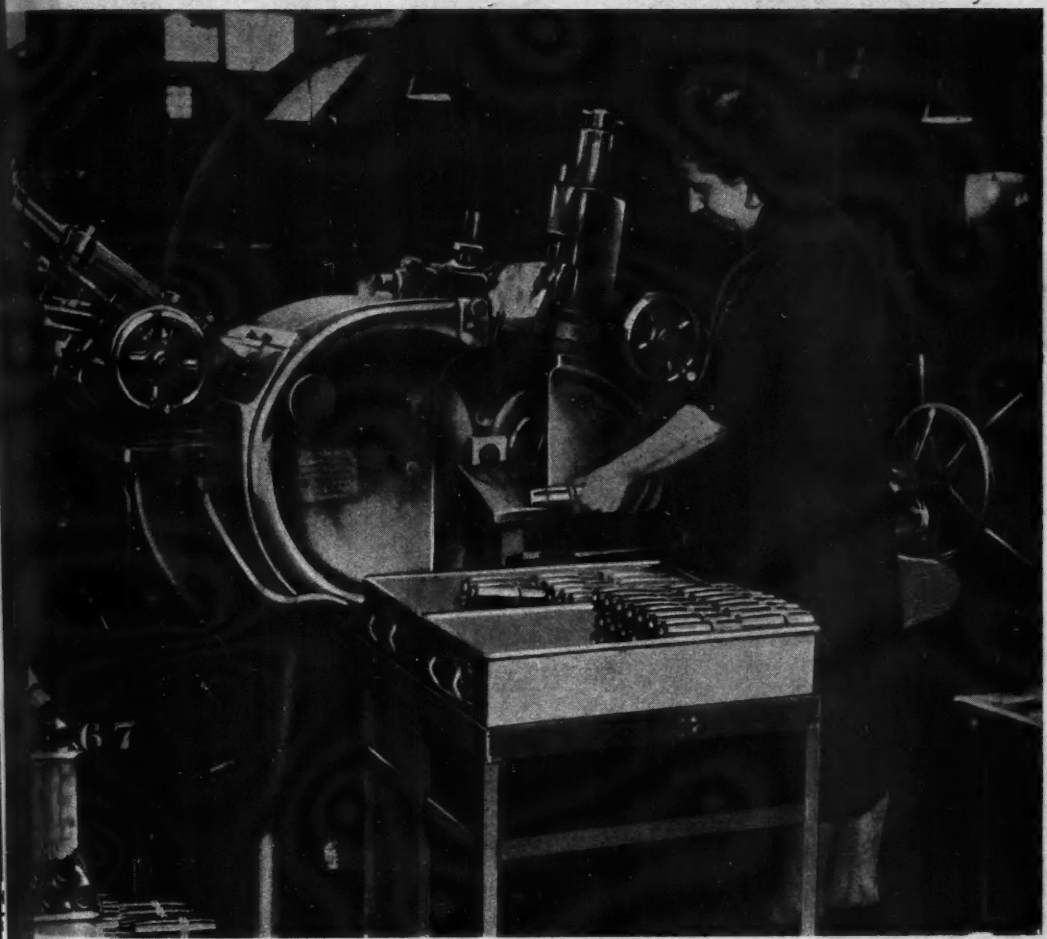

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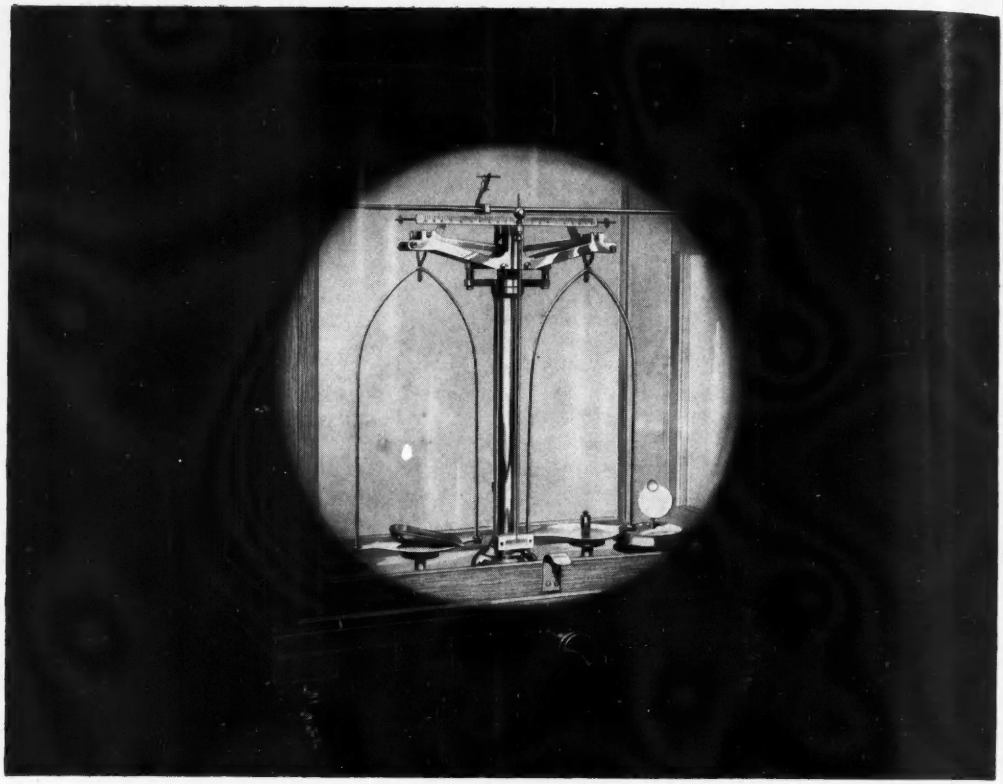

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Editorial

SIMULTANEOUSLY with this issue, the first number of an American edition of " THE ENGINEERS' DIGEST *" will appear in New York. For the first time in the history of British technical journalism, simultaneous editions of a British engineering periodical are published in Great Britain and in the United States, thus forming a link between the engineering communities of the two countries.*

With this purpose in mind, approximately fifty per cent of the editorial contents of the American edition is devoted to Abstracts from Papers of British engineering and scientific societies and from articles published in the current British technical press. Moreover, a number of original articles dealing with industrial developments in this country have been included. We would like to place on record, as well as to gratefully acknowledge, that engineering societies and the editors of all British technical papers whom we approached, have most generously consented to our publishing abstracts from their papers and articles.

In the United States, the response to the project of an American edition of " THE ENGINEERS' DIGEST *" has been most gratifying. Mr. Harold V. Coes, President of The American Society of Mechanical Engineers, kindly wrote a message of welcome in the first number. Engineering concerns of world-wide repute have reserved advertising space to manifest their support of our scheme, and a very large number of technicians, engineering firms and technical libraries have already placed yearly subscription orders with us, prior to publication.*

In normal times the supreme value of an engineering periodical lies in the publication of articles of an educational character, of use in modern research and developments, which can either be of direct value to its readers, or give them the stimulus to investigate further the requirements of their own particular work in view of the trend of scientific and industrial progress. In war-time however, the primary aim of an engineering journal must be to assist its readers in their concentrated efforts to design and manufacture ships, vehicles, guns, bombs, aeroplanes and other instruments of war ; to plan for the most economic employment of material and labour ; to stimulate the invention of devices which may lead to shortening the war, thus hastening the beneficial consequences of an early victory.

Engineers have a particularly eminent part to play in the present war. Their great share in achieving ultimate victory and in securing a world which will be free from Nazism and Fascism, must not be forgotten when peace returns.

The writer for many years past has urged that engineers should be given a greater influence in directing and controlling world affairs. Engineers' brains are trained to calculate the sequence of operations well beforehand, and none are more qualified than members of the engineering profession to be put in charge of the most complicated mechanism of world reconstruction. If we want to prevent the recurrence of this conflagration through which we are now fighting our way successfully, and if we want a stable world with the possibility of progress and prosperity for all, engineers must bring their knowledge and experience to bear upon measures to be taken.

International relations have, however, become so complicated that close co-operation of engineers in all parts of the world is imperative. Close co-operation and agreement between the engineering communities of Great Britain and the United States—brought about by

consultations of Joint Committees, which should include the engineering societies representing all branches of engineering—may constitute the nucleus of a World Confederation of Engineering Societies. Such a World Confederation, representing about a million graduated and qualified engineers all over the globe, would have a decided influence over world affairs.

The war has brought British and American engineers in a much closer association, and it is intended and hoped that the exchange of information through " THE ENGINEERS' DIGEST *" will contribute to a still better understanding between engineers on both sides of the Atlantic.*

It would be most difficult to imagine what the world would be like without the technical and scientific contributions of Davy, Faraday, Watt, Thomson, Perkin, Clerk Maxwell, Stephenson, Armstrong, Royce, Bidwell, Hughes, to mention only a few of the many outstanding British scientific workers and pioneers.

In this connection we must not overlook the contributions made to science and industry by the large number of outstanding engineers, e.g., Edison, the Wright brothers, Joseph Henry, Cyrus Fogg Brackett, Brush, Morse, Conant, Roger Adams, Frank B. Jewett, Freeman, Langley, Carrier, etc., on the other side of the Atlantic.

We believe a better knowledge and appreciation of the scientific and technical work carried out in the two countries would greatly expedite mutual understanding. To achieve this the columns of this journal will always be open to contributions and practical suggestions for eliminating the very great number of divergencies which exist at present in technical developments, measurements, standardisation, even in technical terms and the many other implements of the engineering profession which could, with no great difficulty, be unified to the advantage of engineers in both countries and the world in general.

The venture of launching an American edition of a British technical journal in wartime has been an arduous task. It would have been impossible without the valuable assistance of Mr. K. Headlam-Morley, Secretary of The Iron and Steel Institute; Mr. J. E. Montgomery, of the Institution of Mechanical Engineers; and Mr. J. G. Crowther, of the British Council, whose friendly support has helped us to overcome the initial difficulties. We acknowledge their help with sincere thanks.

Our thanks are also due to Mr. James McLachlan, General Manager of The Engineers' Digest Ltd., who, after crossing the Atlantic, established our Company in New York and has built up the organization of the journal in the United States.

We must also express our thanks and appreciation for the efficiency and understanding shown to us in connection with our American project by Messrs. Devine and Willis of the Overseas Trade Department, by Mr. Briggs, Chief Technical Censor of the Ministry of Information, and by the Ministry of Transport.

To all friends here and in the States we are grateful for the support and co-operation afforded us, without which our task would indeed be unenviable.

J. E. PAJZS.

BRITISH-AMERICAN ENGINEERING COLLABORATION

By HUGH P. VOWLES, M.I.Mech.E.

THE year 1776, in which the American colonies issued their Declaration of Independence, was also the year in which James Watt's steam engine was first applied to industry, and Adam Smith's book, *The Wealth of Nations*, was published.

Of all the peoples of the world none were destined to make fuller use of steam power than the inhabitants of North America. Moreover, wealth has been produced in the United States on a scale undreamed of by Adam Smith. Meanwhile, steam power applied to transport has been of cardinal importance in ensuring the continued existence of the U.S.A. as a single entity.

In the early years of the new Republic, the minds of American statesmen were haunted by the seeming impossibility of establishing federal union between communities widely scattered over a vast territory. Washington prayed that his countrymen might have the wisdom to develop America's magnificent inland waterways. Jefferson thought it might not be possible to keep regions so infinitely remote as the Mississipi Valley and the Atlantic Coast in the same union. Few realised at that time the vast potentialities of steam. Erasmus Darwin, it is true, had written in 1793 the prophetic lines :

> " *Soon shall thy arm, unconquered steam, afar*
> *Drag the slow barge, or drive the rapid car ;*
> *Or on wide waving wings expanded bear*
> *The flying chariot through the field of air.*"

But to most men of the time this must have seemed the insubstantial dreaming of a visionary. Yet not to all men. While hardy American pioneers were seeking to overcome heartbreaking obstacles aided only by muscular power and hand tools—men like Ebenezer Zane, who hacked his " trace " from Virginia to Ohio through miles of forest and wilderness—John Fitch was developing plans for his steamboat which made its initial trip in 1786*, to be followed 22 years later by the steamer which Robert Fulton launched on the Hudson. In England other transport schemes were maturing, and by 1825 George Stephenson's locomotive was running over the Stockton and Darlington Railroad. It was the American steamboat and the British locomotive—perfected by the increasingly interrelated efforts of later generations of engineers on both sides of the Atlantic, and reinforced by the development of telegraph and telephone—that solved the problem of uniting States scattered across a continent. Steam power brought hitherto isolated communities into close contact. Men differing widely in outlook were thus enabled to develop views in common, at least so far as this was necessary to create a sentiment in favor of national unity. In consequence hostility to the idea of a central government gradually died away.

It is interesting to look back through the years and note a few examples, taken almost at random, of the many links established between American and British engineers. It will be seen that what at first was no more than a casual interchange of ideas and men, de-developed in course of time into deliberate efforts to encourage British-American engineering collaboration.

As is well known, Britain was for centuries largely dependent on foreign craftsmen for her engineering development. But by 1776 British-born engineers were (as they are still) among the ablest in the world, and it is not surprising that there was a demand for their services on the other side of the Atlantic. It was not that America had no able engineers of her own. But the population (then only 2,500,000) was so small, the territory so vast, and the tasks to be undertaken so forbidding, that assistance from abroad was essential for several decades. Benjamin Franklin had written from London in 1772 to Mayor Rhodes of Philadelphia, strongly urging that experienced English engineers should be engaged on canal development, and the recommendation was adopted for other branches of engineering work as well. For example, David Rittenhouse of Pennsylvania collaborated with an Englishman, Samuel Holland, in the task of surveying and fixing the boundary between New York and Pennsylvania. In 1794 the engineer, Marc Isambard Brunel, was appointed to survey for the canal from Lake Champlain to the Hudson at Albany, and after becoming chief engineer for New York City left for England, where he settled down, and among other notable achievements constructed the Thames Tunnel. In 1800 the Manhattan Company imported an engine from England to supply New York with water, and the English and Scottish engineers associated with the project remained in America, setting up machine shops there. In 1811 Ferdinand R. Hassler was sent to England to procure instruments for making the Coast Survey, instruments which, however, were soon superceded by others made in America. Seven years later the remarkable American inventor, Jacob Perkins, also crossed the Atlantic, taking with him Gideon Fairman, Asa Spencer (inventor of the geometric lathe) and 26 cases of machinery. Whilst in England Perkins made a series of experiments with steam that startled the British engineering world by their daring and novelty, his boiler operating at pressures as high as 1400 lbs. per sq. in. He invented a steam gun, and gave to the Royal Society an account of his experiments demonstrating for the first time the compressibility of water ; whilst the printing firm in which he was a partner produced the first penny postage stamps, turning out in the course of 40 years over twenty-two thousand million British postage stamps by the American inventor's process*.

Engineering contacts between the two countries continued to increase in number, but it was not until half way through the nineteenth century that British engineers began to wake up to developments in America. The Exhibition of 1851 brought to light remarkable examples of American progress, and the British Government sent commissioners to New York to report on the Exhibition it was proposed to hold there in 1853-54. As the Exhibition was not ready on their arrival the Commissioners, who included Joseph Whitworth and George Wallis, decided to make a tour of American industrial districts. A similar tour was arranged by John Anderson of the British Ordnance Department. In reports subsequently made by Whitworth, Wallis and Anderson it was stated that : " The Americans showed an amount of ingenuity, combined with undaunted energy, which we would do well to imitate." In the production of muskets at Springfield it was found that " the parts were so exactly alike that any single part

*" John Fitch," by W. H. Richardson, *Mechanical Engineering*, June, 1932.

*" Jacob Perkins," by H. P. and M. W. Vowles, *Mechanical Engineering*, November, 1931.

will in its place fit any musket," even when made in different years. The mass production of locks, clocks, nails, ploughs and mowing machines was considered equally impressive by the English engineers.

The effect of these reports was reinforced by other reports on a factory established in England by Samuel Colt, whose American automatic machinery was employed for the manufacture of pistols. The factory was visited by James Nasmyth, inventor of the steam hammer, also by John Anderson. " It was impossible " said Anderson "to go through that works without coming out a better engineer." Nasmyth confessed himself humiliated by the experience, adding : " The acquaintance with correct principles has been carried out in a fearless and masterly manner, and they have been pushed to their full extent ; and the result is the attainment of perfection and economy such as I have never seen before."

With steady improvement in facilities for travel and communication, contact between engineers in the two countries became more frequent, sometimes resulting in direct collaboration. The laying of the first Atlantic cable (1858-1866) was naturally an event of considerable importance in this regard. It made the name of William Thomson—later Lord Kelvin—as well-known in America as in England. When in 1890 a commission of experts was selected to advise the American promoters of the Niagara power scheme, Thomson and Professor W. C. Unwin were among those appointed commissioners.

From the beginning of the present century numerous opportunities for collaboration between British and American engineers have been provided by engineering societies and institutions. In 1901 British engineers were able to discuss important questions with representatives from America visiting the International Engineering Congress at Glasgow. Again, in 1904, the Summer Meeting of the Institution of Mechanical Engineers was held in Chicago in collaboration with the American Society of Mechanical Engineers. At about the same time, on the invitation of the American Institute of Electrical Engineers, a number of British and American electrical engineers met at St. Louis, where discussions took place on electro-magnetic units and international standardisation.

A little later, in 1906, the Institution of Electrical Engineers made arrangements to welcome guests from overseas, including members of the American Institute of Electrical Engineers and the Canadian Electrical Association. In 1910 a Joint Summer Meeting in London was attended by members of the American Society of Mechanical Engineers, when papers on railway electrification were read and jointly discussed.

Other notable meetings of British and American Engineers took place in 1932, and more recently at the Third World Power Conference and Second Congress of the International Commission on Large Dams held in Washington in 1936. An elaborate programme for British-American collaboration was prepared for 1939 by the American Society of Mechanical Engineers and the Englineering Institute of Canada, including a British-American Engineering Congress in New York in which the Institution of Civil Engineers would have participated, together with the Institution of Mechanical Engineers, the American Society of Civil Engineers, the American Society of Mechanical Engineers and the Engineering Institute of Canada. Unfortunately, as a result of the international situation, this magnificent opportunity for British-American engineering collaboration had to be abandoned.

American and British engineers have worked together on the International Steam Tables Committee, also in the investigation of properties of fuels for internal combustion engines.

Among numerous engineering developments involving collaboration between the U.S.A. and Canada one of the most important is the project for developing the St. Lawrence Waterway, which is expected to effect huge reductions in transportation costs by enabling the ocean shipping of the world to reach the heart of the continent, and may provide power resources of the order of 5,000,000 kW.

The Newcomen Society for the Study of the History of Technology furnishes an example of engineering collaboration on another plane, that of historical research ; a subject in which engineers have only latterly evinced an interest. The Society, with an international platform, was founded in 1920 as an outcome of the James Watt Centenary Celebrations of 1919. In 1923 membership was spread to New York by personal intercourse. A visit to America of the Honorary Secretary of the Society had as its aim the definite object of making the Society more widely known there. Although in European countries progress has always been slow, and has lately been interrupted and inhibited by the war, there are now altogether nearly 3,000 members of the Society ; those in the U.S.A. being spread over 43 States of the Union. The American members take a keen interest in the Society, which, besides promoting the study of technological history, is undoubtedly rendering valuable service in fostering international co-operation and goodwill.

It has already been indicated that engineers played a significant part in ensuring closer contact between the constitutent States of the U.S.A. During the century and a half that has elapsed since the Declaration of Independence, American and British engineers have also abridged the distance between the Old World and the New. Whereas it formerly took several months to cross the Atlantic, it can now be crossed in a matter of hours. Thus it is apparent that engineers are bringing the peoples of the world closer together, rendering possible between nations the peaceful collaboration that has been established in the U.S.A. between one State and another.

It is true that engineers also provide means of destruction. But engineers are by nature creative. They are builders and have no wish to destroy. Modern warfare is not caused by engineers, but by failure to adjust the cultural and economic superstructure of civilisation in conformity with improvements made by scientists and engineers in the material foundations of society.

Despite the present world conflict it is not unreasonable to believe that the necessary adjustments will be effected, and so make a repetition of such conflict impossible. In this connection it may be noted that problems of social adjustment can be far more readily solved in an atmosphere of international goodwill. American and British engineers in particular can contribute effectively to the establishment of world peace by working even more closely together in friendly collaboration.

THERMODYNAMIC POSSIBILITIES OF A COMPOUND DIESEL

By Professor DR.-ING. J. L MANSA. (From *Ingenioren, Maskinteknik, Copenhagen,* 7th August, 1943, pp. 89-95).

IN the Büchi supercharging system the gain achieved in output is derived from the diesel engine alone, while in the Götaverken system, it is the low pressure turbine alone which is responsible for the increased output. Considering these facts, it should be possible to develop a "Compound Diesel Engine" in which supplementary power is furnished by both the diesel and the low pressure units. Such an engine can be built in various ways. But this investigation is solely concerned with the Compound Diesel cycle, which is remarkable for the fact that it can be realised with the use of well-known engine components.

The compound cycle is as follow: Atmospheric air is compressed in a supercharger to a pressure 12.5 per cent in excess of the commencement pressure p of the diesel cycle, hereby ensuring the pressure margin required for scavenging of the two-stroke cycle engine. The compressor is so dimensioned that 1.3 cu. m. of air is supplied for every cu. m. of air remaining in the cylinder after termination of the scavenging process, and thus serves as combustion air. Referring to Fig. 1, compression begins at the "Receiver Pressure" p and proceeds adiabatically until a pressure of 40 atm. (568 lb. per sq. in.) is reached. The subsequent process of internal combustion is assumed to proceed according to the constant-volume cycle until the pressure has risen to 60 atm. (852 lb. per sq. in.), and after that according to the constant-pressure cycle until combustion is complete. This is followed by adiabatic expansion down to the original volume. Both exhaust and scavenging are assumed to take place at the constant volume.

The exhaust gases, mixed with scavenging air are passed to the exhaust turbine at 8 atm. (113.6 lb. per sq. in.) pressure, the highest permissible gas temperature at the turbine inlet being taken as 650 deg. C. Such a high temperature may not yet be practicable, but it lies, nevertheless, within the range of possibility. Expansion proceeds in the turbine until a final pressure of 1 atm. (14.2 lb. per sq. in) is reached.

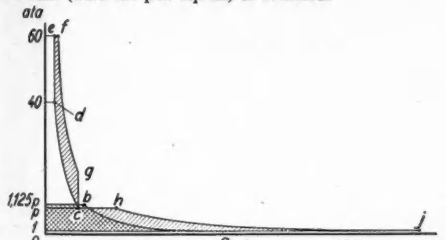


Fig. 1. Pressure-Volume Diagram.

In the computation of the operating cycle of the compound-diesel engine, a compressor efficiency of 90 per cent and a mechanical efficiency of 92 per cent are assumed, based on the employment of a reciprocating compressor. The internal efficiency of the turbine is taken as 85 per cent, and the mechanical efficiency as 95 per cent, the latter including frictional losses in the reduction gear. The ratio of indicated to theoretical cycle output may be 90 per cent; while the heat loss to cooling water may be taken as 20 per cent of the lower calorific value of the fuel.

The mechanical efficiency of the diesel is based on the assumption that the friction losses are equivalent to a loss of 1.0 atm. (14.2 lb. per sq. in) indicated mean effective pressure, thus reducing the computed i.m.e.p. to 14.0-1.0=13.0 atm. (184.6 lb. per sq. in.). Further an excess air rate of 1.75 has been assumed.

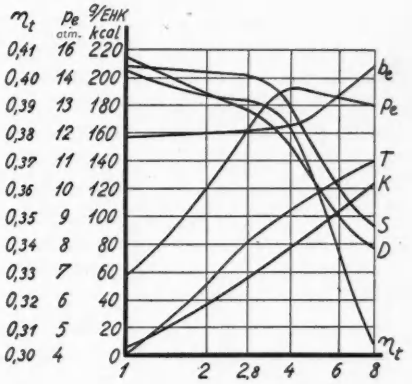
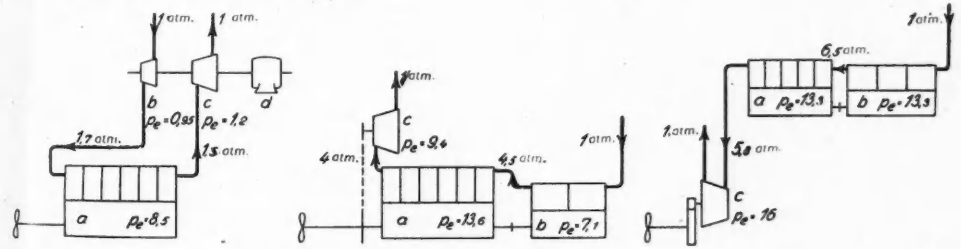


Fig. 2. Computed operating characteristics of Compound Diesel.

D = Useful heat in Diesel, kcal/cu. m. N.T.P. of air.
T = Useful heat in Turbine, kcal/cu. m. N.T.P. of air.
K = Heat consumed in Compressor, kcal/cu. m. N.T.P. of air.
S = Total useful heat, kcal/cu. m. N.T.P. of air.
b_e = Fuel rate, g/h.p. hr.
P_e = m.e.p. in Diesel, atm.
η_t = Overall thermal efficiency based on lower cal. value of fuel.



A: Output of Turbine used for driving of Auxiliaries. b_e = 158 *g/h.p.hr.*

B: Diesel engine, turbine and compressor mechanically coupled. b_e = 165 *g/h.p.hr.*

C: Turbine alone is used as a drive. b_e = 186 *g/h.p.hr.*

Fig. 3. Suggested arrangements of Marine propulsion plant.

Assuming a fuel of 10,000 kcal per kg. (18,000 B.T.U. per lb.) calorific value, the utilisation of the heat supplied is as follows:

Useful heat available in diesel cycle	77 kcal per cu. m. of air at N.T.P.	= 8.655	B.T.U. per cu. ft. of air at N.T.P.
Useful heat available in turbine	140 " " " " "	= 15.74	" " " " "
Total useful heat available	217 " " " " "	= 24.395	" " " " "
Less heat consumption of compressor	124 " " " " "	= 13.938	" " " " "
REMAINING USEFUL HEAT	93 " " " " "	= 10.357	" " " " "
Heat supplied by fuel	306 " " " " "	= 34.394	" " " " "

Thermal efficiency: 93/306 = 0.304 = 30.4 per cent.

By computing the compound diesel cycle for turbine admission pressures ranging from 1-8 atm. (14.2-113.6 lb. per sq. in.), with the admission temperature kept at 600 deg. C., the operating characteristics indicated in Fig. 2 are obtained.

The almost horizontal trend displayed by the efficiency curve η_{th} and the fuel rate b_e for receiver pressures ranging from 1.0 to 4.0 atm. (14.2 to 56.8 lb. per sq. in.) show that within this range the economy of the compound diesel cycle is little below that of an ordinary diesel cycle. But with receiver pressures above 4 atm. (56.8 lb. per sq. in.), the thermal efficiency is seen to drop. The marked increase in mean effective pressure with increasing receiver pressure up to 4 atm. indicates that the compound diesel principle leads to a considerable increase in specific power output as compared to ordinary diesel engines. There is no doubt that this more than twofold increase in output, achieved without a marked increase in the specific fuel consumption, will make the compound diesel a particularly attractive proposition for ship propulsion plants, locomotives, and other applications, where low weight and small space requirements for a given engine output are of paramount importance. Three such compound diesel arrangements, suggested for a ship propulsion plant, are outlined in fig. 3.

The greatest factor of uncertainty encountered in compound-diesel computation is the assumption of the individual component efficiencies to be realised. In evaluating the effect of changes in the component efficiencies, the influence of the cooling water heat loss must not be left out of account. With this point in view, the cycle efficiency has also been computed on the basis of a 10 per cent heat loss due to cooling water. But in this case, the lower calorific value of the mixture must be decreased to 260 kcal. per cu. m. N.T.P. (29.22 B.T.U. per cu. ft. N.T.P.) as compared to the originally assumed value of 300 kcal. per cu. m. N.T.P. (33.72 B.T.U. per cu. ft. N.T.P.) if the turbine admission temperature is to be kept at 643 deg. C. This will result in an overall thermal efficiency of 32.8 per cent, a specific fuel consumption of 193 g per h.p.hr. and a mean effective pressure of 12 atm. (170.4 lb. per sq. in.) as compared to 34 per cent efficiency, 208 g per h.p.hr. and 13 atm. (184.6 lb. per sq. in.) m.e.p. based on 20 per cent heat loss to cooling water.

It cannot, however, be expected that a heat loss to cooling water as little as 10 per cent will be attainable in practice.

POWER CONSUMPTION IN THE ROLLING OF STEEL SHAPES

By M. STEFFES. (From *Stahl und Eisen*, Vol. 63, No. 15, April 15th, 1943, pp. 295-301).

THE kWhr consumption in the rolling of steel shapes is an important factor in establishing manufacturing costs. To be really useful, however, power consumption data must be based upon statistical findings covering a wide range of shapes rolled. The operating capacity of the mill must also be considered. In addition to this, power consumption charts covering the various phases of the rolling process as such must be obtained. These records are essential in investigating questions of roll design and capacity of the mill drive.

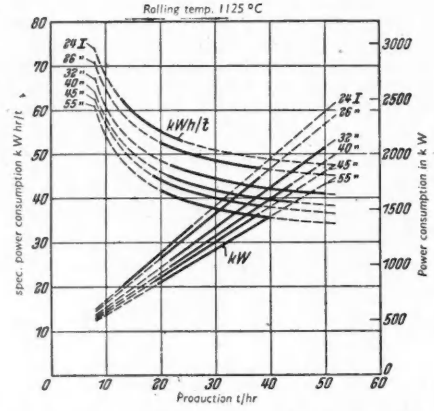


Fig. 1 Power consumption in rolling of I-beams.

The data given in this article are based upon investigations conducted over a period of five years for the purpose of determining the relationship between hourly mill output and shape rolled. They cover 2 structural mills, a merchant mill, and a small rod and wire mill respectively. In order to make the findings of as general a nature as possible, endeavours were made to establish average conditions of mill operation. For this reason only data were used which are based on approximately identical rolling temperature.

All roll stands were equipped with red brass bearings, only the small rod mill having lignum vitae bearings. The kWhr consumption was obtained by meter-reading, while the rolling temperature was determined by optical pyrometer. No power consumption other than that of the mills themselves was taken into account. Fig. 1 shows the power consumption for the rolling of I-beams both as total power consumption and as specific power consumption in kWhr per ton of rolled material. The total power consumption for the various sizes of I-beams is seen to increase in linear relation with the production in tons per hour. Extrapolation of these converging straight lines to the point of zero production yields the power consumption of the mill when running empty. Specially conducted no-load tests have shown these extrapolated values to be in good agreement with actual conditions.

The specific power consumption is seen to fall with increasing mill output and represents a hyperbolic curve. Fig. 1 also shows that for equal production rates, both the total and the specific power consumptions decrease with larger I-beam sizes. Corresponding results obtained with the rolling of channel iron in the intermediate mill train are reproduced in Fig. 2. The

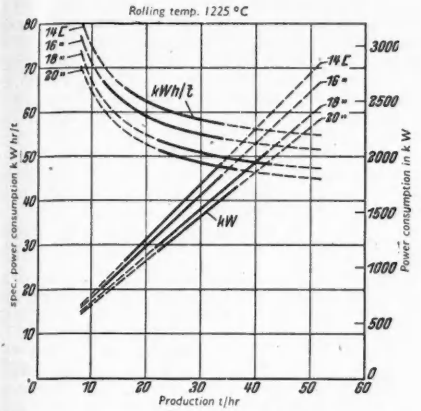


Fig. 2. Power consumption in rolling of channels.

maximum output approximated to 40 tons per hour. When rolling German NP 18 channels weighing 13 lb. per ft., and having a cross section of 4.35 sq. in., the specific power consumption was about 50 kWhr per ton with a mill drive load of approx. 2,000 kW.

Analogous results were obtained with the merchant mill train. In this case, the comparatively high idling power consumption of the shaping section of 100 kW, and a corresponding no-load consumption of 225 kW of the finishing section, leads to a steep increase in power consumption for production rates below 10 tons per hour. Maintenance of high mill capacity is therefore especially important. The separate determination of power consumption with the mill running empty, already referred to, was effected by uncoupling the rolls, and then also the pinions and the flywheel. In this way the rolls were found to take 158 kW, the pinions and fly wheel 45 kW, and the mill drive itself 22 kW. This shows that by far the largest portion of the idling power consumption is due to roll friction.

The results of a special test run conducted with synthetic resin bearings are of particular interest. Comparative power consumption data are given in Fig. 3, where the installation of the non-metallic bearings is seen to result in a reduction in power requirements by 25-30 per cent. as compared to red brass bearings. The rod mill train turns out rounds and wires ranging from 12.7 to 5 mm., using billets 67 mm. square and weighing 660 lb. The power consumption per ton decreases with increased diameter of the material and with the output. When producing 25 tons per hour of 5 mm. diameter wire, the power consumption is 125 kWhr per ton as compared to 65 kWhr per ton when producing the same quantity of 12.7 mm. rods. The empty mill train has a power consumption of 450-650 kW according to the number of stands in operation. There are 18 stands of which at least 12 are operating.

The heat and power consumption balance of the entire plant, including electric generating plant, power distribution system, and rod mill, is given in the subjoined table, which refers to the rolling of 40 tons per hour of 12.7 mm. rods.

	kWhr/ton	B.Th.U's/ton	%
Electricity generation heat supplied	—	900,125	100
Power supplied	60.57	206,800	23.0
El. Distribution losses ..	1.82	6,215	0.7
Mill running empty	11.25	38,410	4.3
Useful mill load	47.5	162,175	18.0

This table shows that based on the total heat supplied to the power plant, only 18 per cent of the potential energy supply is utilised in the rolling process. With the use of spark-type recording instruments, numerous measurements of power consumption and mill drive speed were made. In the rolling of a 4 ton bloom in 15 passes into an I-beam of German NP55 (22″ depth, 8″ flange), the mill drive output curve shown in Fig. 4 was obtained. The use of two ordinates in this graph is explained by the fact that this is a reversing mill. The roll speed in the first pass is small, the roll speeds subsequently mounting to 50-60 revs. per minute and terminating in a max. rolling speed of 80 revs. per minute. The graph also shows working times and idling times, as e.g., an idling time of 7 seconds before the 13th pass, and 3.5 seconds after that pass, the actual rolling time being only 8 seconds. The beam was completed in 15 passes, taking 4 minutes 20 seconds in all.

In analysing the power consumption of the roll mill drive, attention must be paid to the fact that the loss of load at the end of a pass results in an acceleration of the motor and in consequent storage of kinetic energy. But the latter is recovered in the rolling process which follows. Thus, for instance, the actual power consumption in the 13th pass amounts to 10.6

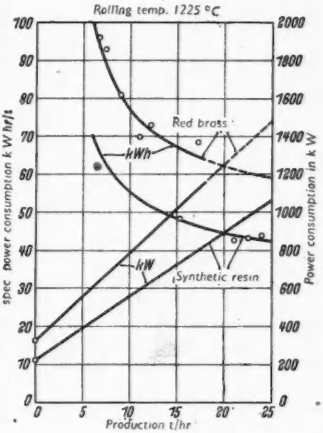


Fig. 3. Power consumption in rolling of channels with metallic and non-metallic bearings.

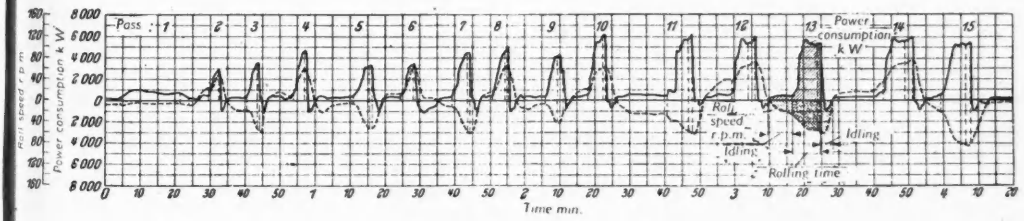


Fig. 4. Power consumption and roll speed of two-high reversing mill train.

kWhr in 18.5 seconds, corresponding to an average power consumption of 2,000 kW with a max. value of 5,340 kW. By totalling up the power consumed in the individual passes, and by basing the total power consumption upon the weight of the bloom as supplied to the mill, a specific power consumption of 22 kWhr per ton by the mill drive is found.

Assuming continuous production, the mill output would be abt. 55 tons per hour, which corresponds to 50 tons per hour of final product. According to Fig. 1, the measured power consumption at this production rate is 34 kWhr per ton. By assuming an Ilgner drive efficiency of 80 per cent, the latter figure will be reduced to 27 kWhr per ton. The difference between the two amounts (27 versus 25 kWhr per ton) is ascribed to inaccuracies in measurement and to differences in production conditions.

RAIL CAR WITH CHARCOAL GAS PRODUCER.

By ING. F. KRALIK, Prague. (From *DMZ, Deutsche Motor-Zeitschrift*, Vol. 20, No. 3, March, 1943, pp. 80-82.)

THIS article describes a rail car now in service on the Bohemia-Moravia Railways which are part of the former Czechoslovak State Railways. The car was designed and built by the Skoda Works in Pilsen.

This new design is not an adaptation to an existing car, but has been developed *ab initio* for this special purpose, as will be seen by reference to the specially designed engine and the location of the producer, etc., in a special compartment at each end of the car (Fig. 4).

The car has two axles, its length over buffers is 12.15 m., width 3 m., empty weight 17,600 kg., maximum speed 65 km./h. There is accommodation for 50 seated and 10 standing passengers. The whole driving mechanism is below floor, only the producer and filters being above floor level at the ends of the car and isolated from the passengers (Fig. 1).

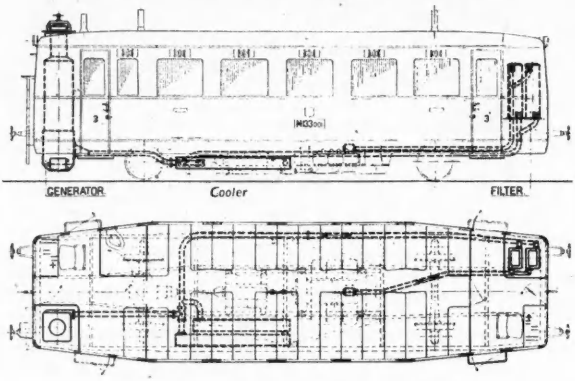


Fig. 1. General layout of the Skoda rail car.

The power unit is a horizontally opposed 12 cylinder engine of 135 h.p. (Fig. 2), driving through clutch and gearbox on to one axle by means of a cardan-jointed shaft (Fig. 3).

The gas producer is of the G.P.-Fiser system, has a rectangular cross-section, and there is no lining.

In the lower part of the producer there are two tuyères supplying air necessary for combustion and generation of gas. They also provide means for igniting the charcoal in the producer. Except for a narrow zone of combustion around the nozzles, the producer is mainly a fuel hopper, in which the fuel itself is a heat insulator of the container walls.

The gas is drawn through a filter and passes a system of coolers and purifiers before entering the engine. The fuel is dry charcoal, so that there is no residue of acids or tar. This fact is largely responsible for simple operation and long engine life. The producer has a capacity for 200 kg. charcoal, corresponding to three hours of running. The consumption is 0.5 kg./h.p.hr.

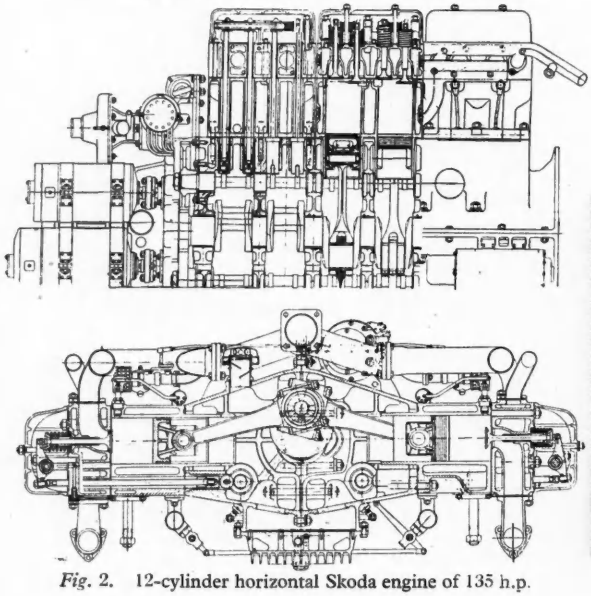


Fig. 2. 12-cylinder horizontal Skoda engine of 135 h.p.

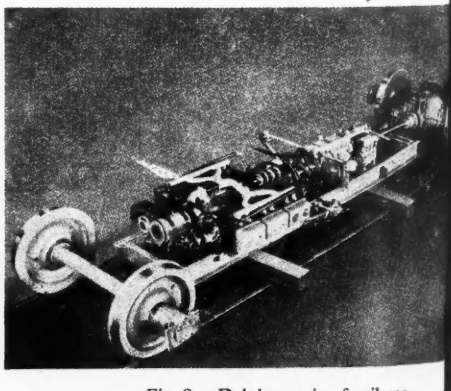


Fig. 3. Driving unit of rail car.

The watercooled four-stroke engine is of 110 mm. bore and 150 mm. stroke. The stroke volume is approximately 17 litres. The engine speed is controlled by a mechanical governor; it is normally 1,600 r.p.m., giving 135 h.p.

The total weight of the engine is 1,130 kg. Its design is clear from Fig. 2. For starting the engine, an electric starter is provided, and the engine starts on gasoline. The driver then ignites the charcoal and, after the engine has started to draw gas from the producer, it is switched over to gas. This occurs after about 10 minutes. After short stops up to 15 minutes the engine can again be started on gas.

Other features are the friction clutch and the four-speed gearbox. The gear change is actuated by hand lever and pull rods. The weight figures show up very favourably. The total tare weight of car, including producer, per passenger seat is 352 kg.

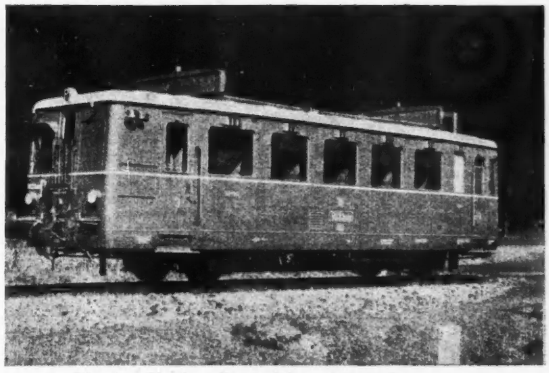


Fig. 4. Skoda rail car driven by producer gas.

HIGH STRENGTH STRUCTURAL STEELS.

By G. P. CONTRACTOR and S. VISVANATHAN. (From the *Quarterly Journal of the Geological, Mining and Metallurgical Society of India*, Vol. 15, No. 2, June, 1943, pp. 69-90.)

RECENT years have witnessed an increasing demand all over the world for high strength weldable structural steels. In addition to strength and weldability requirements, increased resistance to atmospheric corrosion is important. No modern steel, however strong, is considered practical or economical if it does not resist rusting or corrosion. In Great Britain and India the term high-tensile weather-resisting structural steel is applied to those satisfying British Standards Specification No. 548–34, which stipulates a maximum stress of 37-43 tons per sq. in. with a yield point of 23 tons per sq. in. minimum, and a minimum elongation of 18% on 8 in.

On the Continent and in the U.S.A. the tensility range is appreciably lower, being of the order of 33–39 tons per sq. in. with a yield point of 23–25 tons. The higher tensile values required in Great Britain are partly attributable to the tendency of British engineers to base the working stress on the ultimate strength of the material and partly to the fact that the tensile strength of British mild steels is somewhat higher than that of the average Continental and American steels. Continental and American engineers consider that the effective strength of a column depends primarily upon the yield strength of the material. They argue that in most applications the crumpling failure must be guarded against rather than the actual rupture. This tendency has resulted in the production of high-yield strength structural steels calling for an average tensile strength of 35 tons per sq. in. and a yield stress of 25 tons per sq. in. As a result the tensile strength is limited to a maximum of 39 tons per sq. in., but the yield point ratio is increased to something like 75%. This means that the yield point of these 33–39-ton Continental and American steels is about the same as that of the British high-tensile steels. Weldability has also been an important factor which has tended to keep the tensile strength of both American and Continental structural steels at the average value of 35 tons per sq. in. Experience has shown that in fusion welding of low alloy structural steels with tensile strength over 36 tons per sq. in. the parent metal gets weld-hardened. In other words, the zone adjacent to the weld becomes undesirably brittle and may eventually crack in service. Welding difficulties have, therefore, slowed down the application of British high-tensile steels for structural purposes.

From 1920 on there appeared on the American and Continental markets groups of high-strength corrosion-resistant steels with low contents of comparatively cheaper alloying elements. These low-alloy steels are essentially high-yield strength steels with yield point of about 23–25 tons per sq. in. and tensile strength of about 33 tons. In India a similar steel was developed by the Tata Iron & Steel Co., which is known as Tiscor steel. This is essentially a high-yield strength Chromium-Copper-Silicon-Phosphorus steel of low carbon content with remarkable resistance to corrosion. The superiority of this material to ordinary plain carbon structural steel is evident from the data given hereunder :—

	Ordinary Mild Steel (B.S.15)	Tiscor Steel
Tensile strength, tons/sq. in. ..	28 min.	31 min.
Yield point, tons/sq. in. ..	14 approx.	22 approx.
Elongation on 8 in., %	20 min.	20 min.
Resistance to corrosion	—	4 approx.

In spite of its higher strength Tiscor has ample ductility and is easy to fabricate. An important feature is that it can be welded satisfactorily by using any of the several types of coated commercial electrodes. There is no evidence of electrolytic action or accelerated corrosion at the junction of the deposited metal and the parent metal. Tiscor has about 60% greater fatigue strength than the plain steels now in use. Also Tiscor possesses one-third more resistance to abrasion than mild or copper steel. The impact strength of Tiscor compares favorably with that of plain steel at ordinary temperatures, and at sub-zero temperatures it shows marked superiority. It has been shown that 20–25% saving in weight can be effected by the use of this steel. Tiscor is distinctly different from older types of alloy structural steels since it combines low cost with high strength in the as-rolled condition, is practically non-air-hardening, has better resistance to atmospheric corrosion, and neither affects shop practice nor increases fabrication cost.

Results published by British investigators within the last five years emphasise that a low alloy structural steel of 36 tons and more per sq. in. tensile strength has a tendency to develop dangerously brittle weld-hardened zones when welded by ordinary types of mild steel or high-tensile quality electrodes under normal condi-

tions of welding—that is, without preheating the parent metal and without the use of a special welding technique. This unfortunate fact has considerably retarded the application of British high-tensile steels for welded structures. This type, however, was developed in India by the Tata Iron & Steel Co. and is known as Tiscrom. It is essentially a corrosion-resistant, high fatigue limit, chromium-manganese-copper steel with high-tensile strength of 37–43 tons per sq. in. and good ductility. Its safe working stress is approximately 50% higher than that of plain carbon steel. A large tonnage of Tiscrom is employed in the construction of the New Howrah Bridge over the River Hooghly and the Dufferin Bridge near Calcutta.

From a survey of typical Continental and American steels it is concluded that there exist no possible combinations of alloy elements which would satisfactorily and consistently give the tensile strength of 37–43 tons per sq. in. required by British Standard Specification 548, if the carbon content is kept at a maximum low limit of 0.18% in order to avoid the formation of a weld-hardened zone. Dealing with the effects of the various alloying elements available, it is stated that silicon steels possess the serious disadvantage of not lending themselves to welding. High phosphorus steels are sensitive to temperature changes, and welding may cause embrittlement. They are, therefore, dangerous from the welding aspect. High copper content is generally believed not to affect welding quality of steel. Likewise, welding of low carbon chromium steel presents no difficulties. Nickel has a beneficial effect on corrosion resistance, but when going to higher nickel or carbon content with high-strength steels, the factor of weld hardenability is to be considered. Titanium acts to suppress the air hardening tendencies of the steel which give rise to weld hardening.

FRICTION OF WIRE ROPES ON DRIVING PULLEYS

By Prof. Dr. L. Klein, VDI. (From *Zeitschrift d. VDI*, Vol. 86, No. 35/36, 5th September, 1942, p. 554.)

The following summarises results of tests carried out in the Laboratory for Hoisting and Conveying of Hannover Technical College. The friction between wire ropes and the materials used for driving pulleys at present, was investigated for different widths of grooves and for the normally occurring slip velocities. The testing device (Fig. 1) made it possible to record the rope force S_2 for any rope load S_1 and for any slip velocity (Fig. 2). From these values we calculate, as usual,

$$S_1 : S_2 = e^{\mu\alpha}\,;\ \alpha = \pi,\ \mu = \frac{1}{\pi} - \log_{nd}\,(S_1 : S_2).$$

If the spring balance *d* is made to oscillate slightly, the frictional force to change from rest to motion is measured too (S_2R, Fig. 2). All tests showed that friction reaches a maximum for the transition, that it decreases by ap-

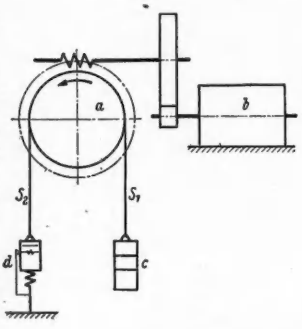


Fig. 1. Device for testing the friction of wire ropes on driving pulleys of various materials.

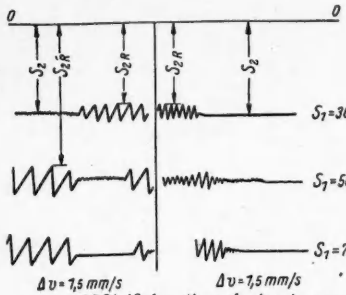


Fig. 2. Reco... the spring ba... for a crossed r... a cast iron g... for various loa... and various ... velocities Δv, ... ρ (radius of g... bottom)=3 ... S_2R=rope for ... transition fro... to movement.

prox. 10% if the slip velocity increases, and that it remains constant from a slip velocity of 6 mm./sec. up to the limits of the tests, i.e., 12 mm./sec. For a slip velocity Δv=6 mm./sec., crossed rope, the following coefficients of friction were found for various pulley materials.

Table I.

Pulley material	Rope condition	Coefficient of friction
Smooth cast iron	Dry or slightly greased	0.13–0.14
Cast iron, 40° V-shaped groove	Dry or slightly greased	0.35–0.40
Greasy leather ..	Slightly greased ..	0.25
	Coated with rope lacquer	0.25
	Coated with sticky lubricant	0.15–0.20
Aluminium ..	Dry	0.8 and above
	Slightly greased ..	0.5–0.7
	Slightly greased with rope lacquer	0.6–0.7
	Slightly greased with sticky rope lubricant	0.3–0.4

As is known, V-shaped grooves give the necessary friction, but rope wear is excessive; therefore, grooved pulleys are not used for more important installations and avoided for small ones as far as possible. The coefficients of leather or textile linings nearly reach the necessary values (μ=0.25 to 0.40 for accelerations of 1 to 2 m./sec.²), but do not attain them, their friction depends very much on grease and humidity conditions, wear is considerable and in case of a stoppage of the hoist cage they may be ignited by the friction of the stationary rope.

At present, only aluminium linings give full security of sufficient friction (DR Patent No. 646611). Manufacturing the pulley rim from aluminium or lining it with this metal is a simple proposition, and has proved to give reliable results. As for wear, there is the example of an aluminium lining which has been worn by 54 mm. after a hoisting output of 614,027 ton-km. useful load, while a leather lining, under similar conditions, has shown an average wear of 52 mm. and a maximum of 67 mm. for 460,633 ton-km. For 1,000 ton-km. this means 0.088 mm. for aluminium and 0.11 to 0.145 mm. for leather. The unequal wear of leather linings may also induce longitudinal oscillations in the rope, which may become dangerous.

The results of the tests as a whole showed that aluminium for driving pulley rims is better, cheaper, simpler and saves the ropes more than any other material used hitherto. Greater rope velocities, and in addition, safe starting and braking, are possible.

STEAM OR GAS-TURBINE?

By DR.-ING. O. MARTIN. (From *Die Wärme*, Vol. 65, No. 49, 5th December, 1942, pp. 419-425.)

WHILE the conception of the gas turbine principle is even older than that of the steam turbine and the internal-combustion engine, numerous investigations and experiments extending over a considerable length of time have made it only too clear that the realisation of this prime mover type is far more difficult than was originally anticipated.

Gas turbine blading was found to be incapable of withstanding the high operating temperatures; the efficiency of the centrifugal compressors proved too low, and the desired thermal efficiency could not be obtained without the use of large heat exchangers. While other novel prime mover principles, such as that of the coal dust engine, lost their original attractions because of the difficulties encountered, the idea of the gas turbine has, however, still retained its vitality. Expenditure on gas turbine development has already far exceeded the amounts spent on the realisation of the diesel principle, and it is still increasing.

Recent reports on gas turbines actually built have given renewed hope of a successful solution of the problem. Significant advances have also been registered in related fields, e.g., exhaust turbine pressure charging of diesels, pressure charging of high-capacity boilers for gas and oil firing, etc. Also an oil turbine driven locomotive has recently been placed on trial service.

The gas turbine must be considered as a fluid-dynamic rather than a thermodynamic problem. Since fluid-dynamic science is still in the development stage, it is difficult to predict what the minimum losses obtainable with blading and heat exchanger of the gas turbine will be. Increased thermal efficiency must, as a rule, be purchased at the expense of increased construction cost. Because of this inter-relationship of physical and monetary aspects, it is difficult to gauge the technical and economic aspects of the gas turbine. For economic reasons the importance of solid fuels is bound to increase in future. The use of gas turbines will, therefore, be confined to a few strictly limited fields, among which operation with blast furnace gas probably ranks foremost.

The suggested use of cooled hollow blading is beginning to lose its original attractiveness since, for various reasons, cooling of the blades has proved to be far less effective than anticipated. While, with the use of lower initial gas temperatures, a great simplification in design and better fluid-dynamic conditions are obtained, this means the abandonment of the hope of achieving commencement of gas expansion at the level achieved with internal-combustion engines. As to the use of high heat-resisting steels, available creep data show that at present their upper temperature limit of applicability lies at about 550° C. This precludes the possibility of constructing gas turbines for admission temperatures of 1,000 °C. or even 1,500 °C.

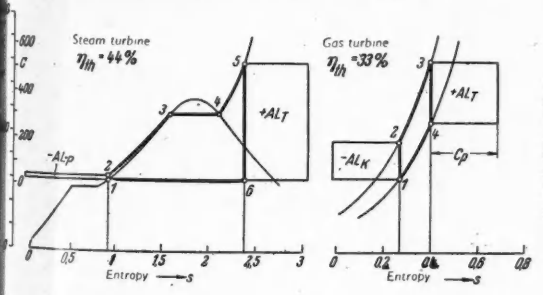


Fig. 1. Temperature-Entropy diagrams of Steam- and Gasturbine.

The inherent differences between the gas turbine cycle and that of the steam turbine are shown in Fig. 1. In the steam cycle the water at condition (1) is put under pressure and thus brought to condition (2), thereby consuming the amount of work L_P. The water is then heated in the economiser and enters the boiler in the condition (3). By superheating, the steam is changed from the saturated state (4) to state (5). During expansion, signified by the vertical (5) — (6), the work L_T is performed in the turbine, the steam-water mixture entering the condenser at (6). The mechanical losses approximately equal the adiabatic work required by the pump and reduce the turbine output by about one-fifth. As even in the high-pressure range the power consumption of the pump is hardly more than 3% of the turbine output, the useful work (neglecting the power consumption of auxiliaries) available at the coupling is

$$L_n = \eta_T L_T - L_P/\eta_P = 0.8\ L_T - 0.03\ L_T = 0.77\ L_T$$

Consequently, the actual turbine must pass 30% more steam than the theoretical one. Furthermore, the power consumption of the pump requires the actual output to be 3% greater than the useful output.

In the gas turbine cycle ambient air of state (1) is compressed to state (2), for which the work $-L_K$ is required. Subsequently condition (3) is attained by heating. After that, expansion in the turbine, making available the amount of work L_T, leads to state (4), in which the gas is released to atmosphere. At the temperatures attainable to-day, the adiabatic work of compression $-L_K$ is about one-half the adiabatic turbine output L_T. In the actual machine, both $-L_K$ and L_T are performed with 85% efficiency. Thus the useful output at the coupling is

$$L_n = \eta_T - L_K/\eta_K = 0.85\ L_T - 0.5\ L_T/0.85 = 0.26\ L_T$$

This shows that the turbine output must be four times the useful load. Three-quarters of the output produced is absorbed by the air compressor and by mechanical losses. The effective thermal efficiency η_{the} of the machine can, therefore, be expressed by $\eta_{the} = \eta_{th} \cdot \eta_M \cdot \eta_K$, where η_{th} is the thermal efficiency of the ideal machine, and η_K is the boiler efficiency. In an internal-combustion turbine, η_K solely comprises the heat losses of the combustion chamber. The overall efficiencies of the two machines are :—

(*a*) For the steam turbine

$$\eta_M = L_n/(L_T - L_P) = 0.77\ L_T/0.985\ L_P = 0.782$$

(*b*) For the gas turbine—

$$\eta_M = L_n/(L_T - L_K) = 0.26\ L_T/(L_T - 0.5\ L_T) = 0.52$$

In spite of the higher machine efficiencies of 85% in the case of the gas turbine set, only 52% of the thermal energy available from the ideal gas turbine cycle is utilised. This compares to a figure of 78.2% in the case of the steam plant, although the latter operates with a turbine efficiency of only 80% and with a pump efficiency of 50%. Compared to the steam turbine, the gas turbine, therefore, suffers from the following fundamental disadvantages :—

(1) The gas turbine cycle requires a much larger turbine, and also a large compressor, the latter being at least as expensive as the turbine itself. Both machines will weigh and cost a multiple of a steam turbine of equal output.

(2) Since the useful output of the gas turbine set is given by the margin of the turbine output over the power consumption of the compressor, the adverse effect of mechanical losses is twofold. Thus, as a matter of principle, in the gas turbine process, the actual output available at the coupling will be a smaller fraction of the output available from the ideal cycle.

Both steam and gas turbine cycle efficiency are still far below that of the Carnot cycle. The regenerative feed heating employed with the steam cycle considerably reduces this shortcoming. Regenerative heating can also be employed with the gas turbine cycle by utilising the exhaust gases for preheating the compressed air charge in a counter-flow heat exchanger of the type shown in Fig. 2. As the power required to overcome the friction drop in this heat exchanger adversely affects cycle efficiency, minimum friction loss is essential. Theoretically, laminar flow will yield higher rates of heat exchange for a given pressure loss than turbulent flow, as the ratio "*Heat transferred/Power equivalent of the friction loss*" increases with the reciprocal of the square of the velocity of flow. But laminar flow requires extremely narrow gas passages. Thus a transfer rate of 1,000 Kcal per sq. m. per hour (368.7 B.Th.U. per sq. ft. per hour) with a mean temperature difference $\Delta\delta = 20$ °C., calls for gas passages of only 1 mm. width. Recuperators of this type, which must be heat resistant, pressure tight and easy to clean, are difficult to design and expensive in construction, as about 3 sq. m. (32.38 sq. ft.) surface are required per kW.

A comparison of the respective capital costs and heat rates of gas turbine, steam plant, and large gas engine, with blast furnace gas as fuel, shows that the gas turbines excel only in cases where very low annual load factors prevail. Owing to the high cost of the recuperator, a gas turbine operating on the regenerative cycle cannot be considered competitive.

In commercial operation the accumulation of dust and soot deposits on the gas turbine blading will have a more deleterious influence upon turbine efficiency than that caused by salt deposits on steam turbine blading. The author concludes that there can scarcely be any hope that the gas turbine will ever attain a success at all comparable to that attending the introduction of the diesel engine.

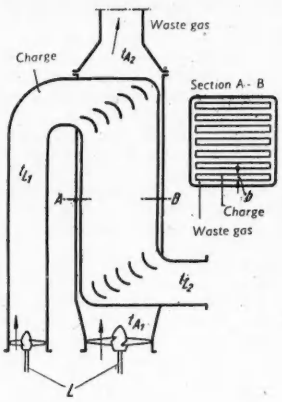


Fig. 2
Heat exchanger of a gas turbine.

ENERGY CONVERSION IN GAS AND OIL TURBINES.

By Professor Dr. Ing. W. Nusselt. (From *Die Wärme*, Vol. 66, No. 15, June, 1943, pp. 139-142.)

This article was written as a reply to the preceding article by O. Martin *on "Steam- or Gas-turbine."*

In the constant pressure process adiabatically compressed air is admitted to a burner which at the same time is supplied with oil fuel. The burner is arranged in the bottom of a combustion chamber which contains the stationary flame. At the outlet end of this chamber the gases flow under pressure to the turbine proper in which they are expanded to atmospheric pressure. A turbine plant of this kind, therefore, consists of air compressor, oil pump, combustion chamber and turbine. Theoretical investigation of this machine is best carried out on the basis of the cycle diagram shown in Fig. 1. This consists of two adiabats and two isobars, the combustion process being taken as external heat supply.

The useful output of this plant is the difference between the power output produced by the turbine and the power consumed by the compressor. The efficiency with which compression and expansion is effected, therefore, is of vital importance. The head diagram of the constant-pressure cycle is shown in Fig. 2. It follows that the thermal efficiency is

$$\eta_t = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{P_1}{P_2}\right)^{\frac{K-1}{K}}$$

Assuming identical compression ratios, the constant pressure and the Otto cycle yield equal efficiencies. As the efficiency of the latter has reached a value of 36% in commercial operation, the prospects of the constant-pressure cycle appear to be highly favourable. Fig. 3, showing the relationship of thermal efficiency and compression ratio, indicates an efficiency of 40% with $p_2 = 6$ atm. (85.2 lb. per sq. in.). As isothermal compression is often believed to result in a higher efficiency than adiabatic compression, this question is investigated on the basis of the diagram (Fig. 4). With isothermal compression the supply of heat commences at point 2′, that is, at the initial temperature T_1, while

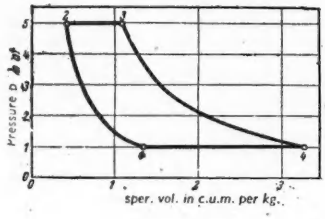


Fig. 1.
Pressure-volume diagram of constant - pressure turbine.

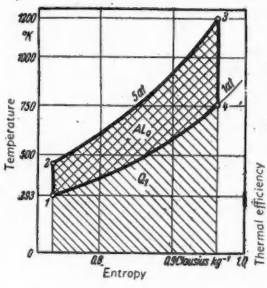


Fig. 2. Heat diagram of constant-pressure turbine.

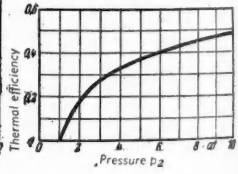
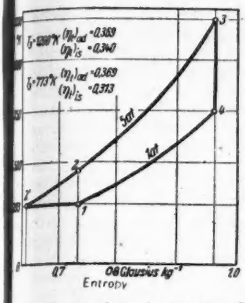


Fig. 3. Thermal efficiency of constant-pressure process.

Fig. 2.

Fig. 3



.4. Comparison between abatic and isothermal compression.

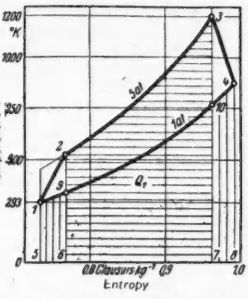


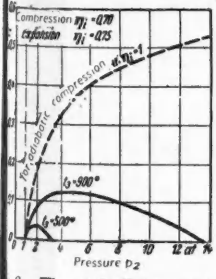
Fig. 5. Heat diagram of constant-pressure turbine with irreversible compression and expansion.

with adiabatic compression it begins with the higher temperature T_2 at point 2. Consequently, the thermal efficiency of a reversible cycle with isothermal compression must be smaller than that obtained with adiabatic compression. Fig. 4 shows that the efficiency differential of the two processes increases with decreasing values of T_3. The situation is, however, different with the irreversible cycle which corresponds more closely to the actual working cycle.

Excepting the combustion process, the most important irreversible process of the actual cycle is the irreversible expansion and compression. Since both compression and expansion must be affected in turbo machines, they are accompanied by a considerable increase in entropy caused by friction in rotor and stationary blading. Fig. 5 shows the constant-pressure cycle with irreversible compression 1—2 and irreversible expansion 3—4, assuming both to take place under friction but without heat exchange. In this case, the useful work AL does not any more equal the area 1-2-3-4 of the cycle but is $AL = Q_1 - Q_2$, that is, it equals the difference of supplied and rejected heat. The supplied heat is represented by the area 2-3-7-6-2 and the rejected heat is given by the area 1-4-8-5-1. The available work is, therefore, equal to the area 2-3-10-9-2, less the area 1-9-6-5-1 and 10-4-8-7-10. The larger the entropy increases 5-6 and 7-8 are during compression and expansion, the smaller will be the work, and with it the thermal efficiency obtained. The indicated efficiency η_i serves as a measure of the irreversibility of the expansion and compression. In the expansion process this indicated efficiency signifies the ratio of the power output at the turbine shaft to the work performed by the adiabatic expansion; while in the stage of compression the indicated efficiency is given by the ratio of the work of adiabatic or isothermal compression to that actually taken up in the compression process. Fig. 6 gives the thermal efficiency *versus* the pressure p_2 at the end of the compression. The curves shown in solid line indicate the efficiency for different maximum temperatures t_3 at the turbine and for irreversible expansion and compression, taking into consideration the indicated efficiencies. But no heat exchange during expansion and compression is taken into account.

While in the reversible cycle the efficiency continually increases with increasing final compression p_2, the introduction of friction causes a fundamental change of the result. In this case the curves attain a maximum at a certain pressure, from which they decline if it is exceeded. In the end the curves return to zero. Thus there exists a comparatively low optimum compression value. Fig. 6 also shows a very pronounced decline in thermal efficiency with decreasing maximum temperature. The lower the maximum temperature t_3 is, the smaller are the optimum compression pressure p_2 and the maximum thermal efficiency.

Fig. 7 gives corresponding data for the case of isothermal compression. A comparison of Figs. 6 and 7 shows that if the friction is taken into account, and if identical maximum temperatures t_3 are assumed, the isothermal compression is more advantageous than the compression with friction, but without heat exchange.



.6. Thermal efficiency of nstant - pressure turbine h compression without heat exchange.

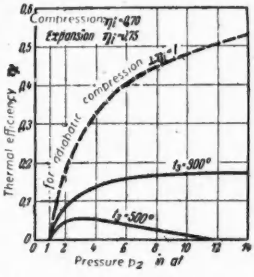
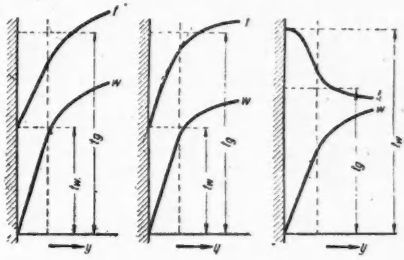

Compression η_i = 0.70
Expansion η_i = 0.75
For adiabatic compression $\eta_i = 1$
t_3 = 900°
t_3 = 500°
Thermal efficiency η_t
Pressure p_2 in at
0 1 2 4 6 8 10 12

Fig. 7. Thermal efficiency of constant - pressure turbine with isothermal compression.



(a) *Fig.* 8. Flow in the boundary layer at the blade surface

(b) *Fig.* 9. Flow in the boundary layer at the blade surface with heat generation in the boundary layer.

(c) *Fig.* 10. Flow in the boundary layer at the blade surface with no heat dissipation from the blade.

It would appear advantageous to use a single-stage turbine of the de Laval type in order to keep the blade temperature as low as possible, as in this case the hot gases, expanded to atmospheric pressure, enter the blading at their lowest temperature. At the high gas velocities prevailing in the blading, the temperature of the latter will not, however, be that of the gas.

In Fig. 8 are shown both temperature and pressure *versus* the distance y on a normal to the blade surface, and it is seen that a linear relationship develops in the

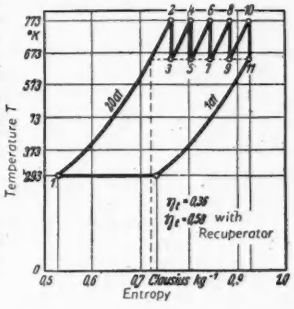


Fig. 11. Heat diagram of Mangold oil turbine with tage combustion

boundary layer on the blade surface. According to Newton's friction law, a shear stress $\tau = \eta$ dw/dy kgm.$^{-2}$ develops in the boundary layer, η representing the viscosity and w the gas velocity. This shear stress is converted into work because of the different velocities at which adjoining particles move. In this way an amount of heat $q = \eta (dw/dy)^2$ kcal. m.$^{-3}$ h^{-1} is generated per unit volume of the gas. Since the velocity gradient increases with the gas velocity, the heat generation in the boundary layer will increase correspondingly, and the parabolic temperature drop shown in Fig. 9 will result. By drawing on findings made by Stodola, F. A. F. Schmidt, Ackermann,* and others, it is shown that this phenomenon causes the blading to assume a temperature which is actually higher than that of the gas, and the temperature distribution shown in Fig. 10 will prevail. Numerical values are given below :—

Gas velocity in m./sec. ..	100	250	500	1,000
Excess of blade temperature over gas temperatures °C. ..	4	27	240	425

Further considering that heat is passed from the boundary layer to both the turbine rotor and the gas stream, it is found that the actual temperature of the blading is smaller than that in the boundary layer, but greater than that of the gas stream. Taking the temperature drop through the de Laval nozzle into account, it is found that the uncooled blade surface attains a temperature only little below that of the gas temperature at the admission side of the stationary nozzles of the turbine.

Use of the regenerative principle increases the thermal efficiency, but heat exchange is only possible if the final temperature of the expansion stage is higher than the final temperature of the compression stage. For identical admission temperatures of the oil turbine, there exists therefore a maximum compression pressure not to be exceeded if the regenerative principle is to be employed. Also, the amount of heat recoverable by regeneration increases with falling final compression pressure. With full regeneration and small compression ratio the efficiency increases to 64%. But this requires an excessively large, and therefore economically not feasible, heat exchanger because of the small mean temperature difference available. An optimum efficiency of 20–26% can be obtained with a final compression to 2 atm. (28.4 lb. per sq. in.) with a temperature difference from 100 — 50 °C. in the recuperator.

Description of the various gas and oil turbines built so far in the course of years is concluded with a discussion of the Mangold turbine, the cycle diagram of which is shown in Fig. 11. In order to keep the blade temperature at 500 °C. the oil fuel is introduced in stages. The turbine is actually composed of a number of turbines arranged in series, sufficient oil being added to the gas flow to reheat it to 500 °C. before it is admitted to the next turbine. Nothing is known about the actual construction of such a unit.

Summarising, it is stated that the efficiency of 18–20% so far attained suffices for many applications. The production of steels resistant to higher temperatures will lead to increased efficiencies. Gas and oil turbines will be used in power plants, and also for locomotives where water is scarce. They will probably also be used for aeroplanes with power plants of 10,000 h.p. and more.

* *Forschung*, No. 6, 1942.

RUBBERLESS TYRES. AN AUSTRALIAN INVENTION

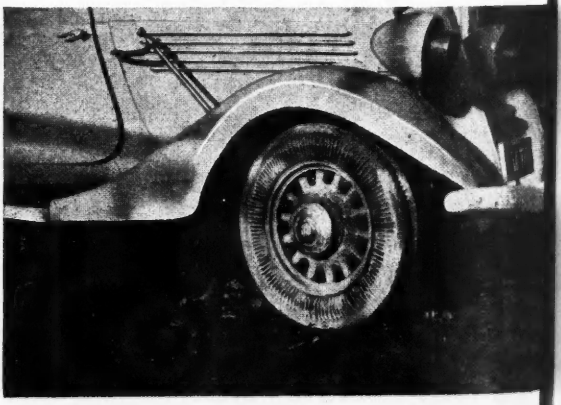
(From *The Commonwealth Engineer*, Vol. 30, No. 12, July, 1943, pp. 294-295).

WITH rubber construction the main duty of the tyre is to transmit the air pressure to the tread so that the load may be sustained and carried with a limited degree of deflection. It is the outer case which provides the essential resistance. If an outer stress is given by air on the rubber case of 120 lb. per linear circumferential inch, then any alternative unit such as the wire case to be described must provide this from the inherent strain on the wire.

Mr. E. G. Stone, of Sydney, states that the tyre which he has developed meets the situation without the use of rubber. It consists of four main features : (1) An outer steel wire casing ; (2) The spacer and structure for the floating ring ; (3) The transmitter and (4) The steel compensator. The outer case, which is spirally wound with wire of a suitable gauge, possesses the necessary resilience, so that the deflection is quite insignificant until the case comes in touch with the transmitter. The transmitter is of felt or other material. This gives under load, but the deflection of the outer case is no more than the wires—upon which the stress is imposed—can sustain. The tread must obviously be one which accommodates itself to present conditions. Felt, or synthetic materials of many kinds may be used. Another element of felt is the dust "extrudor." It is a piece of thin felt, specially treated to resist moisture and to prevent rusting of the wire.

The wires of the case cannot take up the stresses of the full load by their arch action, unless many are brought into play. This is the function of the floating ring. It shortens the span of the arch in the winding ; it carries the stress right around the case and thus gives an equivalent resulting strain to imposed stress that is given by the rubber canvas cover. The floating rings are circular wound small arches which act as continuous beams.

The purpose of the compensator is to take up any stress delivered through the transmitter and the outer case. It is of steel wire of predetermined strength so that if an accident occurs, serious damage to the tyre is prevented. When the pre-calculated stress is reached, and with it the proper deflection, the felt transmitter comes into play and gradually takes up such of the additional movement as is required of it, and when this is exceeded the compensator provides for the remainder. The total deflection of the case and tread is such that no yield point is reached.



EFFICIENT USE OF MATERIALS IN DESIGN BY MEANS OF CORRECT STRESSING

By DIPL. ING. A. ERKER, VDI. (From *Zeitschrift des VDI*, Vol. 86, No. 25-26, 27th June, 1942, pp. 385-395).

THE call for light construction is not solely due to the scarcity of raw materials, but also to the general trend of technical progress. Thus a machine of higher output and lower weight is more economical in many ways. For the realization of light design, there are, in principle, two possibilities open. Firstly, the use of materials of low density, and secondly, improvement in design which enable the designer to come as near as possible to the ideal of stressing, namely, the body of uniform strength. Here all the peculiarities of the individual design such as type of loading, deviation of loading, number of load reversals, etc., must be taken into consideration.

Another aspect of the problem is the maintenance of the shape, i.e., the deformations which oppose the purpose or function of the construction. Thus, for instance, too great elastic deformation with machine tools might have a detrimental effect on the accuracy of the parts to be machined. With other constructions, e.g., structures, elastic deformation is not detrimental; plastic deformation only is to be avoided. Plastic deformation might be due to exceeding the yield point or creep of the material. A third possibility, whereby the original shape of a member might be changed is due to instability. In this case a member suddenly changes its shape at a certain load without rupture, and is still able to transmit forces in the changed shape. The simplest example of instability is the strut under compressive loads (Figs. 1 to 3). Here, however, the instability phenomenon, i.e., buckling of the strut constitutes failure in most practical cases. In other cases of instability, e.g., with stiffened webs, the load carrying capacity subsequent to the occurrence of instability might be even increased.

DESIGN PRINCIPLES

In the following discussion of design principles for various types of loadings, the effect of local stress disturbances, e.g., stress concentration due to sudden decrease of cross-sectional area, will be neglected, and only simple formulas considered.

The notation employed is as follows:—

A	Strain energy [kg. cm.].
d	Diameter [cm.].
E	Young's modulus [kg./cm.^{2}].
e	Distance of extreme fibre from neutral axis [cm.].
F	Area [cm.^{2}].
F_m	Area bounded by mean perimeter of a thin-walled closed section [cm.^{2}].
U_m	Length of perimeter to F_m [cm.].
u	Part of periphery [cm.].
G	Shear modulus [kg./cm.^{2}].
g	Weight per unit length [kg./cm.].
I	Moment of inertia [cm.^{4}].
W	Modulus of section [cm.^{3}].
M	Moment [kg. cm.].
P	Load [kg.].
p	I/e^2F Section coefficient [—].
S	Stiffness [kg., kg.cm.^{2}].
β	Bending angle [—].
γ	Specific weight [kg./cm.^{3}].
δ	Wall thickness [cm.].
ϵ	Strain [—].
σ	Direct stress [kg./cm.^{2}].
τ	Shear stress [kg./cm.^{2}].

Fig. 1. Limited stability.

Fig. 2. Buckling.

Fig. 3. Second stable position.

Tension.

The load which a part under tension is able to carry is solely dependent on the allowable stress (ultimate stress, yield stress, fatigue stress), and the area of cross-section available. By means of a "weight index" the designer is enabled to compare parts as to their respective effectiveness. The smaller this index for a given load P, the greater the saving in weight. For parts under tension the weight index is only dependent on the material used, but independent of the shape of the cross-section.

Table 1 contains the weight indexes of structural parts under tension.

TABLE 1.

Allowable tensile load $P = \sigma F = \sigma \frac{g}{\gamma}$

Weight index for tensile loads: $\frac{g}{P} = \frac{\gamma}{\sigma}$

Stiffness: $S = \frac{P}{\epsilon} = EF = \frac{Eg}{\gamma}$

Weight index for stiffness: $\frac{g}{S} = \frac{\gamma}{E}$

Strain energy in the elastic range:

$$A = \frac{P\epsilon}{2} = \frac{\sigma^2 F}{2E} = \frac{\sigma^2 g}{2\gamma E}$$

Weight index for A: $\frac{g}{A} = \frac{2\gamma E}{\sigma^2} \cong \frac{\gamma E}{\sigma^2}$

Compression.

The same principles apply here as to tension, with the exception that the allowable load for some materials is different from that in tension (e.g., wood, cast iron).

In addition, the instability phenomena which is not present in tension must be considered.

A slender strut under compression will buckle at a certain load, but after removal of the load immediately at the onset of buckling the strut regains its initial shape. Therefore, we speak of elastic buckling (Fig. 4). For a given material and length the buckling load depends on the moment of inertia of the section. Thus, by increasing the radius of cross-section the buckling load is increased. This leads to the employment of tubular struts where excess weight is saved by using thin walls, and yet capable of carrying higher loads.

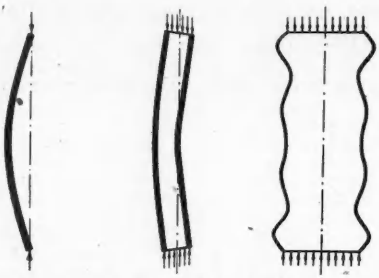


Fig. 4. Buckling in the elastic range. *Fig.* 5. Buckling in the plastic range *Fig.* 6. Local buckling.

The criterion of strength in this case is the yield strength of the material. Accordingly, we speak of plastic buckling (Fig. 5). Further increase of the tube diameter permits still greater reduction in the wall thickness. A limit is set, however, to this process, as the wall may buckle before the strut as a whole buckles. We call this phenomenon local buckling (Fig. 6).

Weight indexes for the described conditions are given in Table 2. Again, the smaller the weight index, the greater the weight saving.

TABLE 2.

1. Plastic buckling: Yielding $P_F = \sigma_F F = \sigma_F \frac{g}{\gamma}$

   Weight index: $\frac{g}{P_F} = \frac{\gamma}{\sigma_F}$

2. Elastic buckling: $P_K \cong E I = E p e^2 F = E p e^2 \frac{g}{\gamma}$

   Weight index: $\frac{g}{P_K} \cong \frac{\gamma}{p e^2 E}$

3. Local buckling: $P_B \cong \delta^2 E \cong \frac{g \delta}{\gamma d} E$

   Weight index: $\frac{g}{P_B} \cong \frac{\gamma d}{E \delta}$

From the weight indexes curves can be developed which show the weight that can be saved by tubular cross-section as compared with the full section (Fig. 7).

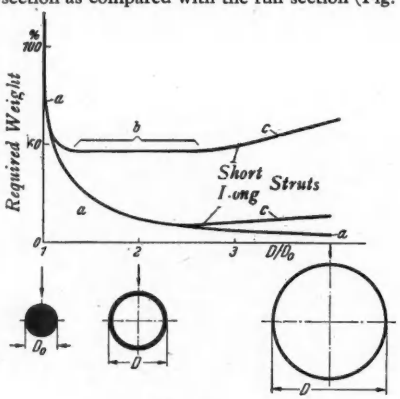
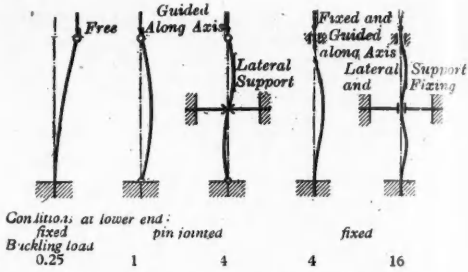


Fig. 7.
(*a*) Buckling in the elastic range; (*b*) Buckling in the plastic range; (*c*) Local buckling.



Figs. 8-12. Increase in buckling load according to Euler, by means of guides and fixing.

The lowest weight obtained is always at the limiting value between over-all buckling and local buckling. By various types of support (Figs. 8–12) the buckling load can be increased despite the reduction of total weight.

Bending.

In bending the influence of the shape of the cross-section is of primary importance, as the modulus of section W depends on it.

As Table 3 shows, W can be expressed as the product of the section coefficient, distance of extreme fibre from neutral axis and cross-sectional area. This clearly indicates the influence of the shape of the section. The more so with the weight index for stiffness where e appears to the second power. It is remarkable that in the expression of the weight index for strain energy, e does not appear at all. This means that if e is large, the bending moment is large and the deformation small. However, the same strain energy can be produced if e is small by a small bending moment and a large deflection.

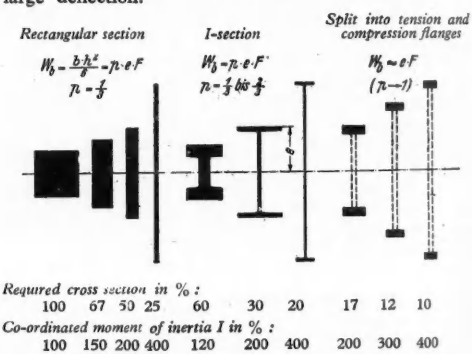
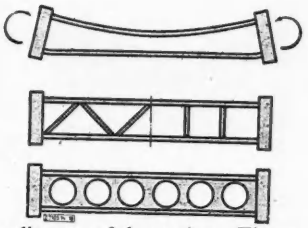

Rectangular section
I-section
Split into tension and compression flanges
Required cross section in % :
100 67 50 25 60 30 20 17 12 10
Co-ordinated moment of inertia I in % :
100 150 200 400 120 200 400 200 300 400

Fig. 13. Comparison of sections of constant modulus W_b.

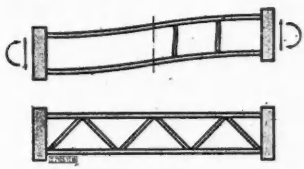
Fig. 13 illustrates a weight comparison for various sections, which is effected in accordance with the formulas in Table 3. The most favourable from the weight point of view is the complete splitting of the section into a compression flange and a tension flange. This, however, might lead to instability of the compression member (Fig. 14). The danger of instability can be avoided by bracing uprights or diagonals between compression and tension members (Fig. 15), which is the framework type of truss. The same advantage can be achieved by webs with holes for lightness (Fig. 16).

While in pure bending the splitting of the section is of advantage, additional shear forces might reverse this effect. Considering the deformations of the split section beam in Fig. 17, one finds that not the total section modulus is employed, but only the sum of the



Figs. 14-16. Light construction of beams under pure bending.

moduli of the two split parts of the section. The uprights are of little value, as they have practically no effect on the deformation. Diagonals have to be used which transmit shear forces (Fig. 18). Full, thin webs can also be used with advantage.



Figs. 17 & 18. Light construction of beams under bending and shear forces.

TABLE 3.

Allowable Bending Moment: $M = \sigma W = \sigma p e F = \sigma p e \frac{g}{\gamma}$

$$\left[\text{For Walls } M_w \cong \sigma e^2 = \sigma e \frac{g_w}{\gamma} \cong \sqrt{\sigma}\frac{g_w}{\gamma}, \text{ as } e \cong \frac{l}{\sqrt{\sigma}}\right]$$

Weight index for bending moment: $\frac{g}{M} = \frac{\gamma}{\sigma p e}$

$$\left[\text{Weight index for walls: } \frac{g_w}{M_w} \cong \frac{\gamma}{\sqrt{\sigma}}\right]$$

Stiffness $S = EI = E p e^2 F = E p e^2 \frac{g}{\gamma}$

$$\left[\text{For walls: } S_w \cong E e^3 = E e^2 \frac{g_w}{\gamma} = \sqrt[3]{E}\frac{g_w}{\gamma}, \text{ as } e \cong \frac{l}{\sqrt[3]{E}}\right]$$

Weight index for stiffness: $\frac{g}{S} = \frac{\gamma}{E p e^2}$

$$\left[\text{Weight index for walls: } \frac{g_w}{S_w} \cong \frac{\gamma}{\sqrt[3]{E}}\right]$$

Strain energy in the elastic range:

$$A = M\beta \cong \frac{\sigma^2 W^2}{EI} = \frac{\sigma^2 p F}{E} = \frac{\sigma^2 p g}{E\gamma}$$

Weight index for strain energy: $\frac{g}{A} = \frac{\gamma E}{\sigma^2 p}$

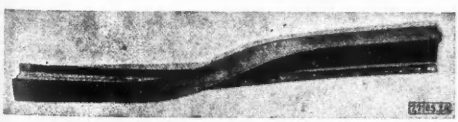


Fig. 19. Sideways buckling of an I-beam.

Narrow, high sections of low torsional stiffness and low resistance against lateral loads are liable to further types of instability, called sideways buckling. This is a phenomenon whereby the beam tends to bend in the plane of least moment of resistance (Fig. 19). This buckling sideways can be in the elastic as well as the plastic range, according to the length of the beam. To avoid this instability, the torsional stiffness of the section must be increased. This can easily be done by using closed sections. Often, especially with thin-walled structures, corrugations of tubular flanges and of hollow sections too, offer an effective reduction of the danger of instability. Similarly, with walls and floors corrugations can be so placed that their distance from the neutral axis is sufficient to withstand the forces and moments acting perpendicularly to the wall (Figs. 20–22).

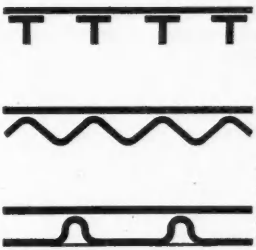


Fig. 20. Simple stiffened wall; the unsymmetrical stiffener does not give an efficient distribution of material.

Fig. 21. Simple wall with corrugated re-inforcement is better, stiffening effect, however, acts in one direction only.

Fig. 22. Correctly designed double wall; almost the whole material is in the highly stressed region.

Torsion.

The weight indexes for torsion are given in Table 4. Again, the smaller the weight index the greater the weight saving, and the table indicates the suitability of closed sections. This is clearly brought out in Fig. 23, where the percentage of cross-sectional area is given for each section.

TABLE 4.

		Full Circular Section.	Closed Circular Tube. $\delta \ll d_m$	Closed Thin-walled Section.	Open Thin-walled Section.
Cross-sectional area F		$\frac{\pi d^2}{4}$	$\psi d_m \delta$	$U_m \delta$	$\Sigma u \delta$
Polar moment of inertia I_p		$\frac{\pi d^4}{32}$	$\frac{\pi}{4} d_m^3 \delta$	$4 F_m^2 \frac{\varrho}{U_m}$ *	$\frac{1}{3}\Sigma u \delta^3$
Polar modulus of section W_p		$\frac{\pi d^3}{16}$	$\frac{\pi}{2} d_m^2 \delta$	$2 F_m \delta$	$\frac{1}{3}\Sigma u \delta^2$ †
Weight Indexes for	Load carrying capacity	$\frac{4\gamma}{\tau d}$	$\frac{2\gamma}{\tau d_m}$	$\frac{\gamma U_m}{2\tau F_m}$	$\frac{3\gamma}{\tau \delta}$
	Stiffness	$\frac{8\gamma}{d^2 G}$	$\frac{4\gamma}{d_m^2 G}$	$\frac{\gamma}{4G}\left(\frac{U_m}{F_m}\right)^2$	$\frac{3\gamma}{\delta^2 G}$
	Strain energy	$\frac{4\gamma G}{\tau^2}$	$\frac{2\gamma G}{\tau^2}$	$\frac{2\gamma G}{\tau^2}$	$\frac{6\gamma G}{\tau^2}$

* The above value is valid for sections without corners (e.g., elliptic tube); for sections with flat sides such as a rectangular tube, it is: $\frac{4F_m^2}{\Sigma u/\delta}$

† The above value is valid with sections of constant wall-thickness; if the section is composed of parts with various thicknesses the value is $\frac{\Sigma u \delta^3}{3\delta_{max}}$

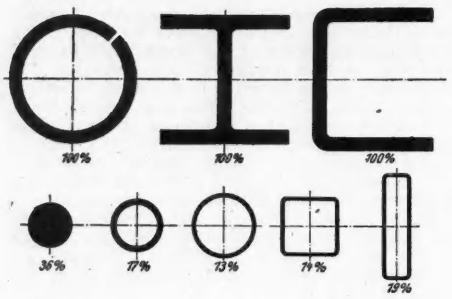


Fig. 23. Sections of equal torsional stiffness.

Deviations from elementary stress analysis

Under this heading mention will be made of subjects with which the author deals more extensively under the titles: notch effect, effect of concentrated loads, fatigue. The subject title, "notch effect" covers the problems of stress concentration due to holes or cut-outs, and the treatment of this part is based on an article by A. Thum and O. Svenson in *Dtsch. Kraftf.-Forsch. Heft* 71, 1942 (*Die Verformungs- und Beanspruchungsverhältnisse an Bauelementen des Fahrzeugbaues*).

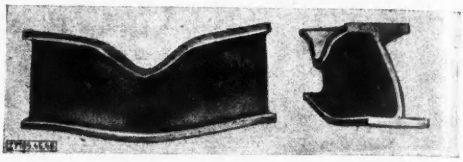


Fig. 24. Instability of I-beams at the application of concentrated loads.

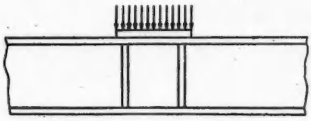


Fig. 25. Prevention of instability by reinforcement and suitable distribution of loads.

The effect of concentrated loads is in some cases identical to that of diffusion, in others it is instability. So, for instance, an I-beam under a concentrated load might fail, not due to too high bending stress on the tension side, but due to instability of the thin web at the application of the load (Fig. 24).

To avoid this the author suggests reinforcement and distribution of the applied load (Fig. 25).

Finally, problems of fatigue and how they affect the choice of materials, are dealt with, and a bibliographical list of 49 papers is appended.

PROGRESS IN GEAR CONSTRUCTION

By H. WITTMANN, VDI., Berlin. (From *Maschinenbau, Der Betrieb*, Vol. 22, No. 1, January, 1943, pp. 9-13.)

Introduction.

IN gear construction Germany must to-day concentrate chiefly on improvements which do not call for new or improved machinery. The aim of such improvements is usually the simplification of design and the ultimate economy of labour and raw materials.

Standardization.

In this connection the use of standardised gear units for a variety of machines and apparatus is one of the first considerations. If this is not possible, standard components should be used whenever possible for production in series or in single units. It is advantageous to introduce manufacturers' standards in addition to the national (DIN) standards. This principle may even warrant overdimensioning of gearboxes or of their parts.

With mass production, on the contrary, every part must be designed for its specific purpose, and even standard parts may be modified.

Materials.

Requirements as regards strength and hardness should be limited as far as possible. Where carbon steels are sufficient, alloy steels should not be used, and the number of different materials in one gear unit should also be restricted. A complete cost calculation is not always necessary to ascertain which of two solutions to a problem is the cheaper one. For two similar gearboxes the one for which the material costs are smaller is the better and cheaper in most cases.

Production.

The production planning department must be furnished with perfect drawings as large as possible which indicate the necessary tolerances, and the required surface qualities must also be indicated. Before starting mass production it is advantageous to carry out tests in order to select the best production methods.

Design.

For the economic production of gearing units, the author proposes to classify them as follows:—

(1) Power gear drives in which the circumferential force attacking the tooth and the circumferential speed are essential for the design.
(2) Control drives in which only a movement is transmitted but no appreciable power, and where only the circumferential speed is of importance.
(3) Uniform-speed drives, for machine tools, measuring apparatus, sighting and similar apparatus to transmit a uniform movement. Besides load and speed, uniformity of angular velocity is important.
(4) Drives for secondary purposes. All those which do not come under any of the above headings.

General Lay-out.

Tooth load and speed must be as high as is compatible with the desired silence in operation. Design should be as simple as possible. As a general rule, planet gears are more expensive to produce and to assemble than equivalent units with lay shafts. Numbers of steps in multi-speed gearboxes should be kept as small as possible.

Spur Gears.

These should be manufactured with standard tools. Number of teeth and width of face should be small; the aversion to a small number of teeth is not justified, as corrections by profile shifting eliminate undercuts.

The basic rack profile according to the 1927 standard (DIN 867) has proved to be good, but in order to

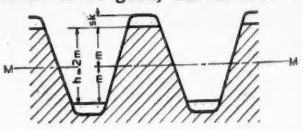


Fig. 1.

Basic rack profile according to DIN 867.

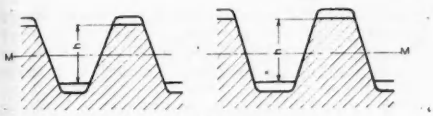


Fig. 2. Basic profile of stub teeth.

increase performance, the author proposes to introduce additional standards for stub teeth (Fig. 1). While the standard profile is characterised by a working depth coefficient

$$y = \frac{h}{2m} = 1$$

the proposed stub teeth (Fig. 2) would have either $y=0.75$ or $y=0.85$. According to gear ratio, the number of teeth on the pinion, and the silence of operation required, one or other of the profiles might be used.

The teeth with $y=1$ have a working depth of $h=2ym=2m$ (Fig. 1). For stub teeth, the working depth would be either 1.7 m. or 1.5 m. The overlap ratio of a gearing indicates how many teeth are engaged simultaneously. With decreasing y this ratio decreases, too (Tables I and II). With increasing overlap ratio, the gears run more silently and their life is increased, all other things being equal. In order not to increase the noise unduly and not to reduce the life of the gears, the ratio is normally kept above 1.4. Where the gears are calculated for only a limited life, overlap ratio may be reduced to 1.3.

As the overlap ratio depends not only on y but also on the number of teeth, stub teeth may only be used above certain minimum numbers of teeth in the case of straight spur gears. For helical gears there is additional overlap due to the helix angle. For stub teeth only uncorrected gears are recommended (zero-gears) or so-called V_0-gears, where the centre distance of the two wheels retains its normal value. For so-called V-gears the pressure angle becomes larger than 20° and the overlap ratio becomes smaller than for O-gears and V_0-gears.

Stub teeth may support higher loads than standard ones, as the tangential force acts on a smaller lever. This is of importance having regard to the present tendency to avoid the grinding of teeth as far as possible. The power needed for cutting stub teeth is less than for standard ones.

A New Arc-Tooth Gearing System.

For all gearings, special emphasis is laid on the absence of noise. One means of achieving this is by the use of helical teeth, but where these are used axial thrusts are generated, which must be supported by bearings. Herringbone gears are used to eliminate these thrusts, but are not very simple to produce if both halves are to bear equally. If axial fixation is not perfect, noise is produced by axial movements of the wheels. The arc gearing system replaces the arrow of the herringbone gears by an arc; it is more silent than herringbone gearing and production is simpler. The newly developed machine for cutting so-called Forster gears is more efficient than the Sykes machine used for herringbone gears.

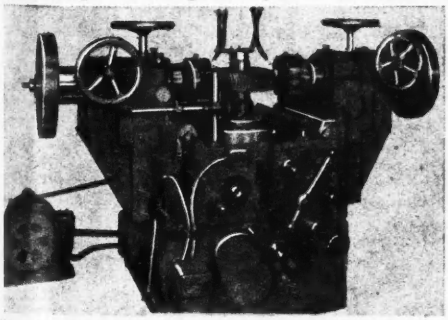
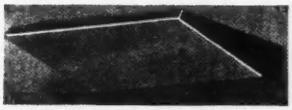


Fig. 3. Forster Gear Cutting Machine. Manufacturers Gebr. Buehler, Uzwil, Switzerland.

Fig. 4.



Cutting blade of the Forster Machine.

TABLE I. Overlap ratios of straight spur gears with profiles according to DIN 867.
Working depth coefficient y=1.

Z_1/Z_2	14	16	18	20	25	30	35	40	50	60
14 ..	1.45									
16 ..	1.45	1.50								
18 ..	1.45	1.51	1.53							
20 ..	1.45	1.53	1.54	1.56						
25 ..	1.45	1.55	1.57	1.58	1.61					
30 ..	1.45	1.58	1.59	1.60	1.63	1.65				
35 ..	1.45	1.59	1.61	1.62	1.65	1.67	1.69			
40 ..	1.45	1.61	1.62	1.63	1.66	1.68	1.70	1.72		
50 ..	1.45	1.63	1.64	1.66	1.68	1.70	1.72	1.73	1.76	
60 ..	1.45	1.64	1.66	1.67	1.70	1.72	1.74	1.75	1.77	1.79
80 ..	1.45	1.64	1.68	1.69	1.72	1.74	1.76	1.77	1.79	1.81
100	1.45	1.64	1.69	1.71	1.73	1.76	1.77	1.79	1.81	1.82
Rack	1.45	1.64	1.75	1.77	1.80	1.82	1.83	1.85	1.87	1.88

TABLE II. Overlap ratios of straight spur gears with stub teeth.
Working depth coefficients $y=0.75$ and 0.85.

Z/Z	Overlap ratio	
	$y=0.75$	$y=0.85$
14/14	1.16	1.28
16/16	1.18	1.31
18/18	1.20	1.33
20/20	1.22	1.35
25/25	1.26	1.40
30/30	1.29	1.43
35/35	1.32	1.46

Production.

Forster gears have teeth which are shaped to cyclic curves; they have involute profiles. For cutting, the machine, Fig. 3, is used, which works on the generating principle. In addition to a continuous feed motion of the two tool supports, there are only constant-speed rotary motions. After the end of the feed motion the wheel is finished. The generating motion is common to the teeth gear and to the tool.

The machine has six change gears, used according to module and number of teeth. The dividing gear is

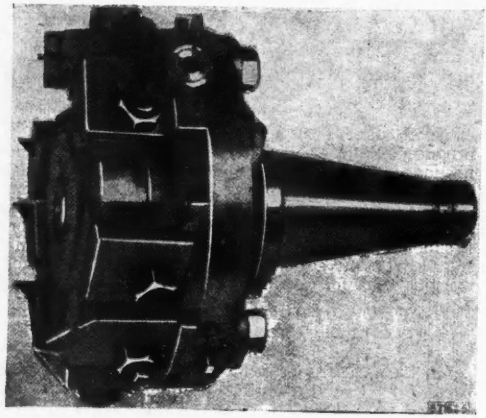


Fig. 5. Cutter head of the Forster Gear Cutting Machine.

an integral part of the machine, and is used for all numbers of teeth. As there are no reciprocating masses, neither foundations nor fixation to the ground is necessary. On the machines built so far, wheels may be produced up to 400 mm. diameter and 120 mm. face width.

The tools are prismatic cutters (Fig. 4) gripped in the cutter head (Fig. 5). Manufacture and dressing of these steels is very simple. Cutting and rake angle may be varied as required by the working material. The steels may also be provided with carbide tips. In each milling head there are roughing and finishing steels, which come to work alternatively. They are independent of the pressure angle and of the module. It is no longer necessary to adhere to the DIN 760 standard modules.

As the Forster gears produce no axial thrust, bearing arrangement in the gearbox is quite simple. Flank pressures may be higher, as contact of the curved flanks is more intimate. Cutting speeds are higher than standard as there are no vibrations in the machine.

When assembling gear units with these gears, the gear faces of a pair in mesh must be perfectly aligned to ensure good contact of the flanks. The influence of deformation at hardening upon the teeth is still to be investigated.

COMPRESSED GAS FOR MOTOR VEHICLES

By DIPL. ING. E. SOENS, Strombeek (Belgium). (From *Brennstoff- und Wärmewirtschaft* Vol. 25, No. 3, March, 1943, pp. 45-49).

State of Development up to 1938.

Although the gas technique in Belgium is well-known to be very advanced, with the exception of a few sporadic experiments no practical experience was available with compressed gas driven vehicles until the end of 1938. At this time, two gas-tank stations were erected near Brussels, one by the firm La Louviere for town gas, and the other by the firm Chimiques du Marly for methane gas (furnace gas). The calorific value of town gas is about 4,200 Kcal/m³, and that of methane gas about 7,500 Kcal/m³. At the beginning of 1939 eleven cars were put into service by Chimiques du Marly using bottled methane gas. The average daily mileage of these cars was 78 miles, including about 120 stops. The firm La Louviere had at the same time about 10 vehicles in service.

In France 9 tank stations were in existence in 1936, all of them in northern France. A few hundred vehicles were converted to bottled gas operation, but further progress until 1938 was very moderate.

In Berlin experiments were made on a much larger scale with omnibusses on a very frequented route. In a few months time over 1,200,000 miles were covered by these vehicles and all the results carefully noted. In order not to inconvenience passengers, it was essential to use interchangeable metallic gas bottles which were changed at the end of the bus routes. The bottles were of 100 litres capacity each and contained gas under a pressure of 350 kg/cm².

These experiments came to an abrupt end in 1938 due to a number of accidents following bursting of the bottles. To prevent further accidents of this nature, all bottles more than one year old were ordered to be replaced with new ones, since it was suspected, and later verified, that the chief reason for the bursting was due to corrosive impurities and small quantities of water contained in the town gas. Another interesting observation was that no accidents occurred with bottles in which methane gas was stored. All town gas tank stations were naturally closed down and the general development of utilizing compressed gas was practically paralysed in Germany. At this period more attention was focussed on liquified gas.

Compressed Gas Tank Stations.

Tank stations must be erected at carefully selected distances apart. The stations must be equipped with a gas compressor with a delivery pressure of 350 kg/cm², and one or more steel bottles of 1000 litre capacity for reserve. It is advisable to have the compressor in two units, in case one of them fails. The author suggests that much more attention should be given to cleaning of the gas before admission into the compressor. This would constitute a precaution against accidents such as have occurred in Germany.

To-day there are over 80 tank stations in service in Belgium, with a total capacity of 13,000 m³ gas per hour. Of this 4,100 m³/h is for methane gas and the rest for town gas. Tank stations are provided 50 km apart along the main routes.

The present turnover of these stations averages for about 120 to 300 m³/hour so that it is estimated that some 200 cars are operating on methane gas and 100 to 120 on town gas.

Auxiliary Equipment needed for the Vehicle.

(A) Bottles. The capacity of the bottles is generally standardized to 10 m³ gas at 200 kg/cm² pressure. The weight of the bottles varies between 65 kg and 100 kg depending upon the material used. These bottles are proof against accidents, they are tested under 300 kg/cm² and burst only under 600 kg/cm². A number of tests were made with these bottles to determine their safety and it was found that even roughest handling such as might occur during a smash up will only result in bulge in the bottles. When the bottles were fired at from a rifle, as an example, the gas escaped through a small hole cut in the metal, but did not cause any damage. Compressed gas driven cars have been in road accidents subsequently, but in no case did the gas bottles add to the damage.

(B) Pressure Regulator. The pressure regulator reduces the gas pressure to a few mm water gauge under atmospheric from the pressure in the bottles which varies from zero to 200 kg/cm². The functioning of the pressure regulator must be very sensitive, partly not to permit leakage of gas from the bottle at say 200

kg/cm², and partly to supply the engine at a few mm water gauge vacuum. It is naturally important that the regulation be fully automatic. That this is quite a feasible requirement is apparent from the fact that one of the test engines has run over 80,000 km without any failure of the pressure regulator. The gas at the required pressure is admitted to the choke tube of a standard carburettor so that the engine can be changed over to petrol fuel without any serious engine modification. With gas fuel, it is well-known that higher compression ratios can be tolerated, i.e., of the order of about 1:8.5 to 1:10.

In order to save fuel by cutting down the extra distance to be travelled to gas-tank stations, movable tank stations were designed, which carry larger containers of 500 litre capacity at 350 kg/cm². An average of 30 cars are supplied every day by one of these stations. The author recommends this method of fuel saving where a large number of cars are to be regularly supplied with gas.

Pros and Contras for Compressed Gas.

Table I. gives some data for petrol, methane and town gas. With methane the decrease in power is about 15%, while with town gas the decrease is about 20-25% compared to gasoline. Some interesting results were obtained by Prof. Coppens on a F.N. engine. On this engine, using town gas and under ideal conditions, the deduction of power was 17½%, which supports the figure given in Table I. The tests showed

TABLE I.

Fuel	Calorific value	Calorific value of gas-air mixture	Power decrease compared with gasoline	Consumption with 4-ton vehicle	Number of bottles with 20% saving; 100 km.
	Kcal/m³	Kcal/m³	%		
Petrol	7500	910	—	28–32 l.	—
Methane gas	7500	820	10–15	30–35m³	4
Town gas	4200	720	20–25	60–65m³	8

that the loss of power was higher with new engines than with relatively older engines. This is due to the better utilization of petrol in a new engine. On the other hand recent improvement of carburettors have resulted in an increase of petrol supply at heavy loads, while with compressed gas the gas supply depends only upon the pressure drop in the choke tube (venturi). As Dr. Rixman has pointed out, this means that no adjustment of one gas regulator will give good results at both partial and full load.

Dr. Rixman has shown that maximum power with town gas is developed with a 5% rich mixture, while at partial load the most economic consumption occurs with 20% excess air. Prof. Coppens' experiments have fully supported these results. Table II. shows the maximum power and corresponding consumption figures for petrol and gas. The ratio of maximum power with gas/max. power with petrol = 0.825. The corresponding ratio of fuel consumption is for gas/petrol $\frac{0.739}{0.401} = 1.82$. When the carburettor was set to give optimum economy, this ratio became 0.693/0.356 = 1.95 m³ per litre of petrol. These tests clearly indicate that no carburettor setting will simultaneously deliver maximum power and reduce consumption to a minimum.

TABLE II.

Fuel	r.p.m.	Mean pressure per cm²	H.P.	Consumption H.P. hour	Loss of power %	Conform. Consumption
Petrol	2000	6.25	18.44	0.401 l.	—	1 l.
Town gas	2000	5.18	15.26	0.739m³	17.5	1.82m³

The consumption figures in Table I., being representative of actual engine operation, show that compared to petrol 1.12 m³ methane gas and 2 to 2.2 m³ town gas can replace 1 litre of petrol.

To calculate the gas quantity actually delivered to the tank the following equation can be used:

$$Q = a\,V\,(p_2 - p_1),$$

where Q is the actual quantity of gas, p_1 initial gas pressure, p_2 final gas pressure in the bottle, V the capacity of bottle and "a" a coefficient of delivery. This coefficient is of considerable importance as variations in consumption of about 20% are obtained, depending whether its true value is considered or whether it is taken as a = 1. The value of the coefficient depends upon the heating effect during filling the bottles to a high pressure.

Considering the overall costs per mileage, and taking into account also the loss in useful load-carrying capacity due to the weight of bottles, the author shows that there is no saving in using town gas in conjunction with heavy (100 kg) gas bottles. For methane gas with heavy or lighter types of bottles there is a saving compared with petrol amounting to about 25-30%.

THE PRODUCTION OF ALUMINIUM FROM BLAST FURNACE SLAGS

By WILHELM E. KREBS. (From *Stahl und Eisen*, Vol. 62, No. 35, 27th August, 1942, p. 735.)

THERE are several methods of obtaining pure bauxite (Al_2O_3) which is the necessary starting material for the electrolysis of aluminium. If the chemical and physical properties of this bauxite reach the required standard the material is finely ground, roasted with soda, and bleached. The aluminium is then precipitated. Hard bauxite with high contents of silica and iron oxide should be used with lime, when the resulting calcium-aluminium compounds will be found to be soluble.

For re-melting, the Miguet furnace is found to be most suitable. This is an electric shaft furnace in which the bauxite is mixed with burned lime and coke. The melt is periodically tapped. Iron and silica occur as by-products and a sow of high melting point remains in the furnace and must be broken out from time to time. The slag often contains titanium.

Re-melting of bauxite has also been carried out in iron blast furnaces, based upon experiments made by Russian investigators.

Ural bauxite has the following composition: 40–45% Al_2O_3, 15–20% Fe_2O_3, 4–10% SiO_2.

Using this bauxite, a series of experiments was made in a blast furnace of 930 m³ capacity at Tula. From January to September, 1939, approximately 100,000 tons of slag resulted from the smelting of 90,000 tons of crude iron. Standard coke containing 11.8% ash and 1.7% sulphur, is not favourable. Better results were obtained from the use of old Makejewka coke containing 7% ash and 1% sulphur.

Some considerable difficulty and many breakdowns were experienced as a result of the extreme viscosity of the slag. In spite of the addition of dolomite lime to decrease the melting point and hence viscosity, the slag continued to cause disturbances in the base

of the furnace and the experiments were discontinued.

The temperature of the crude iron was 1480-1500°C. and that of the slag 1590–1670°C. These temperatures were measured by means of optical instruments. The mixture of ores and fluxes comprised 5 tons of coke, 3.5 tons bauxite, 3.4 tons limestone, 0.15 tons manganese ore, 4.5 tons steel scrap.

In the light of the observations made at Tula a series of experiments was started at Zaporozhe early in 1940 in a blast furnace of 450 m³. Melted Ural bauxite was mixed with Jelenowka lime and Zaporozhe coke scrap in the following proportions :—

8.8 tons coke, 9.5 tons bauxite, 7.5 tons steel scrap, 7 tons limestone.

These experiments met with varying success until June, 1940, when they were terminated by the bursting of the blast furnace. This furnace was closed in September, 1939, after 5½ years' working resulting in an output of 1,626.000 tons crude iron. Operations were resumed after only 45 days, slag being left behind in the furnace, and cracks merely superficially filled in instead of being radically repaired. The very high temperature necessary for the melting of bauxite eventually dissolved the slag, and the crude iron immediately penetrated the cracks and cascaded from the furnace to a distance of 20 m.

It is evident that if a blast furnace is to be used for melting bauxite it must be constructed of a refractory material suited to the chemical and physical properties of bauxite and its slags.

In another blast furnace of 480 cm capacity at Dnjepropetrowsk, bauxite was melted without difficulty and with very good results from July, 1940, until June, 1941. With the outbreak of the war the delivery of Ural bauxite was interrupted. The notebooks of Russian engineers gave the following particulars :—

Mixture of Ores and Fluxes.

Coke (9–12% ash)	5.0 tons.
Bauxite	3.5 tons.
Limestone (1–15%)	3.0 tons.
Steel shavings	3.0 tons.
Blast pressure	1.3 atm.
Blast temperature	750–800°C.
Coke	1400–1500 kg/t. crude iron.

Composition of Crude Iron.

Carbon	4.5–5%	Manganese	0.8–1%
Silica	2–3%	Phosphorus	0.1–0.15%
Sulphur	0.01–0.05%	Titanium	0.6–1.2%

Composition of Slag.

Alumina	45%	Iron oxide	2%
Silica	7.2%	Sulphur	1.6%
Lime	44%	Titanium oxide	0.4%

Crude iron was converted to pig and used for castings. The slag was tapped into pans and cast into specially prepared moulds of 0.5–0.7 m thickness, from which it was removed by dredges after 1½–2 weeks. Finishing was done at the aluminium works at Zaporozhe by breaking, grinding and bleaching.

Laboratory experiments have shown that of the various lime-bauxite compounds CaO, Al_2O_3 has the highest solubility. $5CaO.3\ Al_2O_3$ also bleaches well, but all other components are almost insoluble. In practice the mixtures of ores should be such that the slags have the following composition :—

5.5% silica, 42–43% lime, 49–50% bauxite.

Should the silica content drop below 5.5%, magnesium oxide in quantities of 2.3% must be added to obviate excessive viscosity of slag. The latter should cool in layers at least 0.5 m thick, and the process should be extremely slow.

The experiments emphasize the importance of the following points :—

In general blast furnaces of medium size are most suitable. Constant silica content of the mixture, the use of a coke, low in ash, and regular tapping of iron and slag are conditions essential for success. All frame masonry must be built of the best Chamotte stone. Temperatures must not be allowed to exceed the point required by the viscosity of the slag in use.

NATURAL FREQUENCY OF OSCILLATION OF SPRING MOUNTED MACHINES.

By DIPL.-ING. K. DE GRUBEN, Berlin. (From *VDI- Zeitschrift*, Vol. 6, No. 41-42, 17th October, 1942, pp. 633-637.)

To prevent the transmission of vibration produced by a machine to the foundation or to the supporting structure and/or to protect the machine against vibration, flexible elements are fitted between the machine and the supports. This forms an oscillating system, and it is a well-known principle that resonance between periodical disturbances generated by the machine and the natural frequency of oscillation of the system must be avoided.

An oscillating system in space has six different natural frequencies. The vibrations are in general torsional vibrations which may be partially combined. The direction of the natural vibrations depends on the three moments of inertia of the system with regard to the three principal axes and on the arrangement of the flexible elements. If the system has a vertical plane of symmetry with regards to the masses as well as to the flexible elements, oscillations may occur in this plane the axes of which are normal to this plane. In the present article, the author considers only systems possessing such a vertical plane of symmetry which encloses the longitudinal X axis and the vertical Z axis. In many machine constructions, and especially in I.C. engines, such is very nearly the case.

Further restrictions are not made ; different types of springs may be used in any desired arrangement. In most cases, however, they will be so arranged that the "centre of gravity of springing"* is situated in the vertical axis of the masses, the Z-axis. In this case the system carries out a vertical vibration along a straight line, the so-called fundamental vibration, of a frequency $\omega_o = \sqrt{c/m}$, where c = spring stiffness or restoring force per unit displacement in the vertical direction and m = mass of the system.

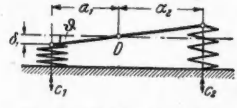


Fig. 1. Definition of the restoring moment.

The spring c_1 is compressed by an amount $\delta_1 = a_1\delta$. At the free end of the spring a force $\delta_1 c_1$ acts and produces a moment $\delta_1 c_1 a_1 = \delta c_1 a_1^2$ about the axis D. For a unit angle of twist the restoring moment is $c_{M1} = c_1 a_1^2$.

The Restoring Moment.

The damping effect of the springs considered is small, and can, therefore, be ignored. In this case the differential equation for torsional oscillation is

$$I_D \varphi + c_M \varphi = 0 \qquad \ldots \qquad \ldots \quad (1)$$

with I_D = polar moment of inertia about the axis, D, of the torsional oscillations, and c_M = restoring couple of the spring, i.e., the couple needed to turn the system by a unit angle against the spring resistance.

* This term will be explained more fully below.

The restoring moment of a spring of stiffness c_1 and situated at a distance a from the axis D (Fig. 1), is $c_1 a_1^2$. The restoring moment of a system composed of several springs of constants c_1, c_2, etc., situated at distances a_1, a_2, etc., respectively from the axis of rotation is,

$$C_M = \Sigma C_i a_i^2 \quad .. \quad .. \quad .. \quad (2)$$

The oscillation considered in the following deflects the free end of a spring usually not only in the direction of its own axis, but also in the direction of a straight line which encloses with the principal axes the angles α_x, α_y, α_z respectively. Since the amplitude of oscillations are always small, the deflections of the free end can be regarded as linear, also in case of a torsional oscillation. The resistance of a spring against displacement of its free end in an arbitrary direction can be expressed by the restoring forces along the three axes. To mark the difference, the notation *d* is used for the spring stiffness acting in a direction normal to the spring axis. Hence, if the free end of a spring is deflected in an arbitary direction, the resulting restoring force is,

$$c_{res} = c \cos^2 \alpha_z + d_x \cos^2 \alpha_x + d_y \cos^2 \alpha_y \quad (3)$$

and the corresponding restoring of a spring, the free end of which is at a distance *r* from the axis is,

$$c_M = r^2 (c \cos^2 \alpha_z + d_x \cos^2 \alpha_x + d_y \cos^2 \alpha_y) \quad .. \quad (4)$$

If the axis of twisting is parallel to the transverse or Y-axis, equation (4) becomes,

$$c_M = ca^2 + d\,b^2 \quad .. \quad .. \quad (5)$$

and for this case, the total restoring moment of a system of springs becomes

$$C_M = \Sigma c_i a_i^2 + \Sigma d_i b_i^2 \quad .. \quad .. \quad (6)$$

For helical steel springs *d* has the same value for deflections in any direction, and its value can be calculated, or better still, experimentally determined.

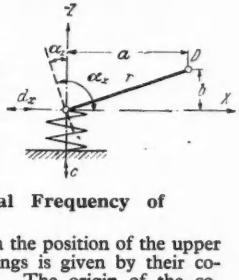


Fig. 2

Calculation of the restoring moment c_M in the axis of torsion D is parallel to the Y-axis and not at the height of the free end of the spring.

Equation for the Natural Frequency of Oscillation.

In the following paragraph the position of the upper free ends of the various springs is given by their co-ordinates s_i and h (Fig. 3). The origin of the co-ordinates is so selected that it coincides with the centre of gravity S of the body. Using these notations, we may also define the centre of gravity of springing. By definition, this is the point the abscissæ of which is given by $p = \Sigma c_i s_i / \Sigma c_i$ and the ordinate by $q = \Sigma d_i h_i / \Sigma d_i$. The third ordinate is zero, since, in this plane the system is symmetrical. If there are *n* equal springs the co-ordinates of the centre of gravity of springing becomes $p = \Sigma s_i / n$ and $q = \Sigma h_i / n$.

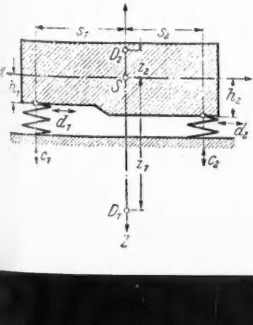


Fig. 3.

Schematic diagram of an oscillating system.

Equations for Simplified Cases.

If p = 0, i.e., the centre of gravity of springing is in the vertical axis of the masses, as is usually the case, one of the six oscillations is linear, and has a frequency $\omega_o = \sqrt{\frac{c}{m}}$, the fundamental frequency. If the springs were all at the same height as the centre of gravity, with q = 0, there would be torsional oscillations around the longitudinal and the transversal axes, and linear vibrations along these two axes. If the springs are not in level with the centre of gravity, but have the respective ordinates, h_1, h_2, etc., the axis of the torsional vibration does not coincide with the C. G. of the body, but is at a certain distance vertically above it. The linear vibration (which may also be considered to be a torsional oscillation around an infinitely distant axis) becomes a torsional oscillation, the axis of which is below the centre of gravity of the body. This is true for both transverse and longitudinal vibrations.

Owing to the original assumption, namely, the symmetry to the XZ plane, the axes of the two longitudinals must be in this plane. The position of the axes of transverse torsional oscillations are thus completely defined by their vertical distances from the centre of gravity. The axis of the longitudinal torsional oscillations need not be horizontal. Strictly speaking, therefore, their angles of inclination should be given, but for practical calculations it is often sufficient to assume that these axes are horizontal. The last of the six oscillations may, therefore, be frequently treated as torsional oscillation around the vertical axis.

The next step is to calculate the frequencies of the torsional oscillations, considering first the transverse oscillations, i.e., oscillations around a transverse axis. If z is the unknown distance of the axis D from the centre of gravity S, the vertical components of the distances of the various springs from the axis of oscillations are $z-h_1$, $z-h_2$, etc., and the expression for the restoring moment (6) becomes

$$\left.\begin{aligned} c_M &= \Sigma c_i s_i^2 + \Sigma d_i (z - h_i)^2 \\ &= \Sigma c_i s_i^2 + z^2 \Sigma d_i - 2z \Sigma d_i h_i + \Sigma d_i h_i^2 \end{aligned}\right\} \quad (7)$$

By using the following abbreviations

$$\left.\begin{aligned} &\Sigma c_i = c. \quad \Sigma d_i = d \\ &\Sigma c_i s_i^2 = cs^2, \quad \Sigma d_i h_i^2 = dh^2 \\ &\Sigma c_i s_i = cp, \quad \Sigma d_i h_i = dq. \end{aligned}\right\} \quad (8)$$

we may rewrite equation (7),

$$c_M = cs^2 + dz^2 - 2\,dqz + dh^2 \quad .. \quad (9)$$

s may be called the effective spring distance and *h* the effective spring height.

If I_Y is the polar moment of intertia of the system about the transverse axis of gravity, and *i*, the corresponding radius of gyration, the moment of inertia about D is given by

$$I_D = I_Y + mz^2 = m(i^2 + z^2) \quad .. \quad (10)$$

From the differential equation of torsional oscillation, equ. 1, the natural frequency of oscillation ω is given by:

$$\omega^2 I_D - c_M = 0 \quad .. \quad .. \quad (11)$$

substituting the values obtained in equations (9) and (10) we get

$$\omega^2 m (i^2 + z^2) - cs^2 - dz^2 + 2dqz - dh^2 = 0 \quad (12)$$

In this equation the frequency ω, and the distance z is still unknown, and therefore a second equation is required. This may be obtained by calculating the restoring couple due to a small rotation of the system and putting their sum equal to zero; the different signs of the forces must be taken into account.

It is of interest to note that this equation may be derived at by partial differentiation of equation (12) with respect to z. According to the rules of differential calculus, this means calculating the extreme values. Thus the axes of rotation are situated in such a way that

rhe frequencies of the corresponding oscillations have extreme values. This is not only true for the special case considered, but it is a general law for vibrations at the natural frequency.

By partial differentiation of (12) with respect to z we obtain the equation

$$\omega^2 mz - dz + dq = 0 \quad .. \quad .. \quad (13)$$

To bring this equation to a dimensionless form, we divide (12) by ci^2 and (13) by ci and introduce some more abbreviations:

$$\left.\begin{array}{l} s/i = S,\ h/i = H,\ q/i = Q,\ p/i = P,\ z/i = Z \\ d/c = C \text{ (" ratio of transverse stiffness ")} \\ \omega^2\ m/c = \omega^2/\omega_0^2 = N^2 \text{ (ratio of frequencies)} \end{array}\right\} (14)$$

and obtain thus,

$$N^2 (I + Z^2) - S^2 - CZ^2 + 2CQZ - CH^2 = 0 \quad .. \quad (15)$$

$$N^2 Z - CZ + CQ = 0 \quad .. \quad .. \quad .. \quad (16)$$

This gives

$$N^2 = \tfrac{1}{2}[(S^2 + CH^2 - C) + \sqrt{(S^2 + CH^2 + C)^2 - 4(CS^2 + C^2Q^2 - C^2H^2)}\ .. \quad (17)$$

$$Z = \frac{I}{2CQ}[(S^2 + CH^2 - C) + \sqrt{(S^2 + CH^2 - C)^2 + 4\,C^2Q^2}] \quad (18)$$

There are two values each for both, the natural frequency and for the distance of the axes from the centre of gravity. The product of the two distances is equal to the square of the radius of gyration.

If all the springs are at the same height, q = h and Q = H, (17) and (18) become

$$N^2 = \tfrac{1}{2}[(S^2 + CH^2 + C) \pm \sqrt{(S^2 + CH^2 + C)^2 - 4\,CS^2}] \quad (19)$$

and

$$Z = \frac{I}{2CH}[(S^2 + CH^2 - C) + \sqrt{(S^2 + CH^2 - C)^2 + 4\,C^2H^2}] \quad (20)$$

In these two equations the two roots have the same values.

For practical applications of the simpler formulæ (19) and (20) are mostly sufficient.

If the springs are all situated at the height of the centre of gravity of the body, h = O and H = O. From equation (19) we obtain

$$\omega_1/\omega_0 = \sqrt{C} \quad .. \quad .. \quad (21)$$

(frequency of linear horizontal vibration) and

$$\omega_2/\omega_0 = s/i \quad .. \quad .. \quad (22)$$

(frequency of torsional oscillation about the horizontal axis of the centre of gravity).

Equations (17) to (22) are also true for oscillations about longitudinal axes, if the corresponding distances of the springs and radii of gyration are introduced.

It remains to calculate the frequency of oscillation around the vertical axis. In most cases, the vertical axis of the system may be considered to be also the axis of the oscillations. If the horizontal co-ordinates of a spring 1 are s_1 and t_1, the distance of that element from the vertical axis is $r_1 = \sqrt{s^2 + t^2}$. The restoring force is determined by the transverse spring constant d_1. The effective distance of the total spring system is accordingly $r = \sqrt{\Sigma d_i r_i^2 / \Sigma d_i}$. Therefore, the frequency of the torsional oscillation around the vertical axis according to (22) is

$$\omega_3/\omega_0 = \sqrt{C} \cdot r/i_z \quad .. \quad .. \quad (23)$$

Influence of the Position of the Springs.

The formulæ show that the horizontal and vertical co-ordinates of the springs have considerable influence upon the natural frequencies. For the case considered in equation (19) a simple graphical representation is possible. Equation (19) may be transformed to

$$\frac{S^2}{N^2} + \frac{H^2}{(N^2 - C)/C} = 1 \quad .. \quad .. \quad (24)$$

This is the equation of an ellipse, which may be easily traced. If $N^2 < C$, the curves become hyperbolæ according to

$$\frac{S^2}{N^2} - \frac{H^2}{(C - N^2)/C} = 1 \quad .. \quad .. \quad (25)$$

Each point of an ellipse, resp. hyperbola, corresponds to a certain pair of values for the co-ordinates *s* and *h*. All spring co-ordinates on a certain ellipse give, for a constant C, the same frequency of the faster torsional oscillation; all spring co-ordinates on a certain hyperbola give the same frequency of the slower torsional oscillation.

Fig. 4 shows for C = 1.0, the two ellipes corresponding to N = 1.2 and N = 1.5. There are also shown two forms of the machine, having the same moment of inertia. For the vertical type (full line) a lower frequency ratio may be attained with laterally arranged springs than with springs arranged below the body. For the horizontal type (dotted line) the lower frequency ratio can not be attained at all as the curves for N = 1.2 are wholly inside the body, where springs may not be arranged or only with great difficulty.

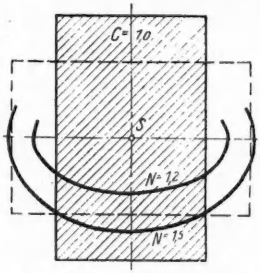


Fig. 4. Influence of the spring co-ordinates upon the frequency ratio N at constant ratio of transverse stiffness C.

The influence of the stiffness ratio C is shown in Fig. 5. With increasing the larger value of C the lateral arrangement of the springs becomes more favorable. With regard to buckling, steel compression springs will seldom have a ratio, C, smaller than 0.5.

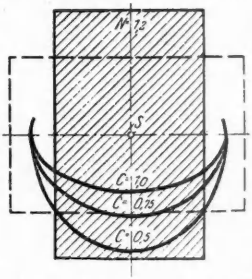


Fig. 5. Relation between spring co-ordinates and frequency ratio N at different ratios of transverse stiffness C.

Only for very low built machine foundations can a low natural frequency be obtained by arranging the springs below the machine. For most machines, and especially for large I.C. engines, lateral springs give better results. This arrangement includes also the so-called " pendulum spring suspension."

Equations for the General Case.

If the centre of gravity of springing is not in the vertical axis of gravity, i.e., if $\frac{\Sigma c_i s_i}{\Sigma c_i} \neq 0$, the axis of the transversal torsional oscillation will not intersect the axis of gravity, but will have the co-ordinates x and z. This case will frequently occur with elastic suspension of engines in vehicles and aircraft. The restoring moment, according to equation (6) becomes

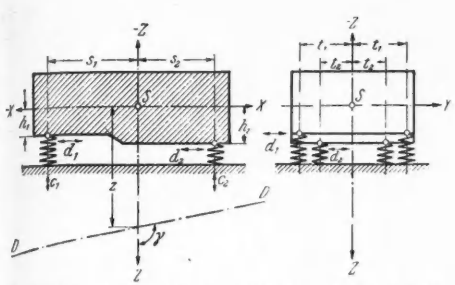


Fig. 6. Torsional oscillations about an inclined axis.

$$C_M = \Sigma (x - s_i)^2 c_i + \Sigma (z - h_i)^2 d_i, \quad .. \quad (26)$$

and the moment of inertia

$$I_D = M (i^2 + x^2 + z^2) \quad .. \quad .. \quad .. \quad (27)$$

Using the abbreviation according to equation (8) and (14), and by partial differentiation with respect to x and z, we obtain two more equations, and finally we get

$$N^6 - (1 + C + S^2 + CH^2) N^4 + (C + S^2 + CH^2 - CS^2 + C^2H^2 - P^2 - C^2Q^2) N^2 - C (S^2 + CH^2 - P^2 - CQ^2) = 0 \quad .. \quad (28)$$

which may be solved by trial and error. One value of N^2 lies between C and 1 ; the oscillation corresponding to this frequency replaces the linear, vertical vibration of the system considered before.

Pendulum Spring Suspension.

For torsional oscillations about inclined axes, e.g., for I.C. engines and large compressors, the so-called pendulum spring suspension has proved to be of great value. The springs are arranged to the two sides of the concrete block approximately at the height of the centre of gravity, (Fig. 6), and are easily accessible from above. The lower ends of the springs are fixed, the upper ends can move freely in all directions. Through bolts are arranged in the springs in such a way that the springs are compressed under the load.

The lower end of the bolts have suitably shaped heads fitting in corresponding holes in the concrete foundation block of the engine so that rotation around the head is possible.

If the foundation is horizontally displaced, the system will execute such a movement which requires the expenditure of least work. It may be proved that the upper end of the spring is compressed by an amount δ_L and displaced horizontally by δ_W; but δ_W is so small compared with δ_L that it may be neglected. This means that the springs are nearly exclusively compressed. The horizontal restoring force of the pendulum spring suspension may be determined with sufficient accuracy from $d_p = \frac{Q}{l}$, where Q is the gross weight and l the length of the through bolts.

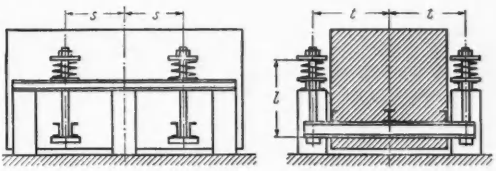


Fig. 7. Pendulum spring suspension.

The frequencies may be calculated according to the methods described above, and for the transversal spring constant the value $d_p = \frac{Q}{l}$ is to be introduced. The vibrations are nearly the same as of a system in which the centre of gravity of springing coincides with the centre of gravity of the machine. There are three straight line vibrations along the main axes with the frequencies $\omega_0 = \sqrt{\frac{c}{m}}$ and $\omega_{1,2} = \sqrt{\frac{dp}{m}}$, and three torsional oscillations having the frequencies $\omega_3 = \frac{W_{ot}}{ix}$

$\omega_4 = \omega_0 \, s \, iy$ and $\omega_z = \sqrt{d_p (t^2 + S^2)/m i_z^2}$.

A special advantage of the pendulum spring suspension is that horizontal linear vibrations and torsional oscillations about the vertical axis have very low frequencies. This is especially important for low-speed single or twin cylinder reciprocating engines. The horizontal impulses originating from the horizontal movements of the masses may be absorbed by the pendulum springs, while the much smaller impulses from the rotating masses are transmitted to stiffer springs.

LOAD TESTS ON BEARING SUPPORTS OF 27,500 kVA HYDRO-ELECTRIC UNITS.

(From *Bulletin Sécheron*, Geneva, No. 14, 1942, pp. 7-10).

WHEN designing a vertical type alternator, it is very important to know beforehand the deflection of the bearing supports. In addition the relatively small clearance between the turbine runner and the guide case forces the designer to strictly limit the increase of deflection when the unit passes from idling to full load. The calculation of the elastic curve for the supports presents no difficulties, in principle. For large units and supports of considerable height, however, the influence of shearing stresses is such that the real elastic curve may differ notably from the one calculated with the use of the bending moments only. The shearing stresses having been considered, it is also important to keep strictly horizontal the central platform on which rest the thrust bearing and the upper journal bearing.

To ascertain the exact behaviour of four support beams destined for the 27,500 kVA hydro-electric units of the Verbois plant on the river Rhone, the firm S. A. Sécheron had them tested in their workshops. The tests were carried out by the Swiss Federal Material Testing Institute (EMPA), under the personal direction of Professor Roš. The supports are in the form of a bridge, consisting of two parallel beams each, and are

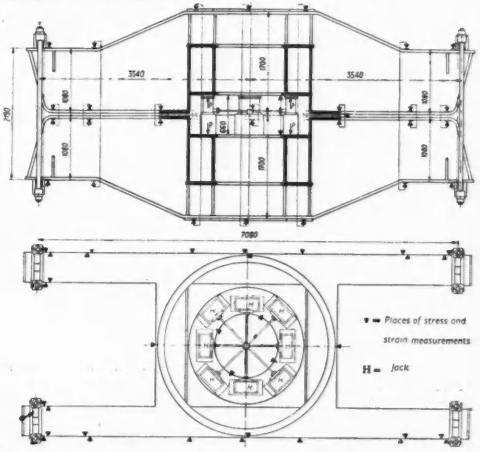


Fig. 1. Disposition of the support bridges for the load tests.

of welded construction. Two pairs of the supports were bolted together, and the test loads applied by eight hydraulic jacks for each pair (Fig. 1). The normal load corresponding to the rated alternator output is 560 tons ; each beam has thus to carry 280 tons. During the tests, the beams were overloaded first by 10% and then by 30%. The full 560 tons consist of the following components : —

Alternator rotor, weight	.. ..	150 t
Turbine runner, weight	.. ..	70 t
Bridge, thrust bearing and exciters	..	60 t
Hydraulic reaction	.. .. ..	280 t

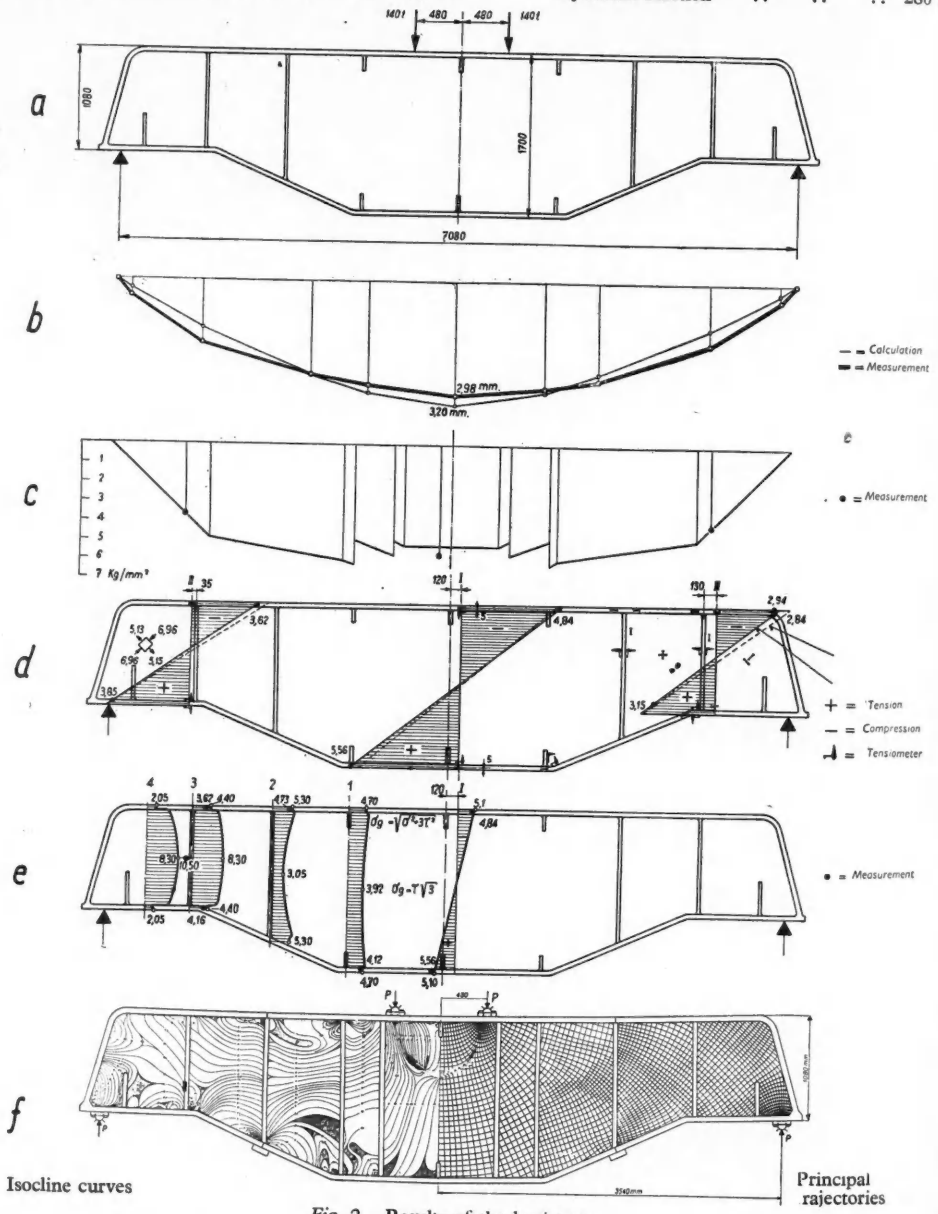


Fig. 2. Results of the load tests.

(a) Elevation of one beam ; (b) Elastic curves (calculated and measured) ; (c) Repartition of tensile stresses along the lower flange ; (d) Calculated and measured tensile stresses in various cross sections ; (e) Static stresses : calculated and measured values of the compound stresses in five cross sections ; (f) Field of the stresses : principal trajectories and isocline curves.

The oil pressure in the jacks attained 560 atm during the tests. Arrangement of the jacks on the central platform corresponds to the real repartition of the thrust at full load.

The elastic curve for 280 t per beam is shown in Fig. 2b, as well as the one calculated from the bending moments and shearing forces. According to Fig. 2b, the maximum deflection is 2.98 mm. The increase of the deflections, very important for the design of the whole unit, from idling to full load, is only 1.52 mm. Fig. 2b shows also the points where stresses and strains were measured. Fig. 2d shows the measured values and the calculated ones, and Fig. 2c shows the repartition of the tensile stresses along the lower flange of the beam. Fig. 2e shows the measured and the calculated strains in five different sections.

For the overload tests, parts of the beams were covered with a varnish calibrated for a yield point of 28 kg/mm². With this lacquer, no flow figures were discovered.

From the tests with the beams and previous tests with the materials used for their construction, the factors of safety were obtained as follows :—

With regard to ultimate tensile strength	4
With regard to yield strength ..	3
With regard to fatigue limit	2

These factors of safety are considered to be sufficient.

BRIGHT ANNEALING OF STEEL STRIP IN ELECTRICALLY HEATED BELL-TYPE FURNACES IN A RUSSIAN STEEL MILL.

By V. A. SOCHINSKI and V. P. GRYAZNOV (From *Stal*, *Moscow*, No 6, June, 1941, pp. 68-70).

In the heat treating department of a Russian steel mill, producing cold rolled strip, bright annealing is effected with the use of 14 bell type electrically heated annealing furnaces. Their number is ultimately to be increased to 30 units, all built by the Ural-Electro-Machine Works. This furnace type is illustrated in Fig. 1. The bell consists of a welded steel housing lined with refractory and heat insulating material.

The lining carries the electric heating elements subdivided in two independently wired zones, each being composed of a number of individual sections. In the furnace type PSK-210, shown in Fig. 1, a third heater zone is provided in the centre of the furnace, but this zone is omitted in the furnace type PSK-180, which is also in use.

Each heating zone is equipped with individual automatic temperature control, the temperature being registered by a recording instrument. The furnace bell is handled by a 15-ton travelling crane. In order to permit continuity of operation, three furnace bases are provided for each furnace bell. The covered work baskets of welded construction containing the charge, are made of ordinary boiler plate of 5-8 mm. thickness.

Positive re-circulation of the protective inert gas throughout the furnace is effected by means of a fan of heat-resisting material centrally arranged beneath the charge. Gas tightness of the furnace is secured by the circular water seal indicated in Fig. 1, the water level being automatically maintained at its proper elevation. These water seals have lately been replaced by oil seals. The protective inert gas used for the bright annealing process is composed of 7-15 per cent H_2 and 85-93 per cent N_2. The gas is admitted through tubing arranged in the base of the furnace, its pressure and rate of feed being controlled by valves in the supply and discharge lines. The temperature in the furnace is measured with a thermocouple connected to a recorder of the potentiometer type.

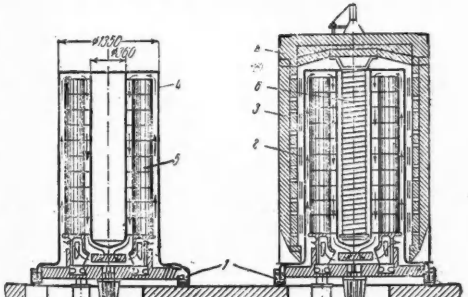


Fig. 1. Bright annealing furnace, type PSK-210.
(1) Water seal (2) Heating element (3) Furnace bell (4) Protective cover (5) Charge (6) Centrally arranged heating element (7) Fan (8) Refractory furnace lining.

The salient data of the PSK-210 furnace are the following:

Rating of the bottom section of the wall heating elements: 90 kW.
Rating of the upper section of the wall heating elements: 90 kW.
Rating of the central heating elements: 30 kW.
Full furnace rating: 210 kW.
Number of phases: 3.
Voltage of wall elements: 380 V.
Voltage of central elements: 220 V.
Number of radiant element section in the furnace wall: 6.
Number of radiant sections in the furnace centre: 1.
Maximum furnace operating temperature: 850 deg. C.
Height of furnace: 3,500 mm. (abt. 11′ 6″).
Outside diameter of bell: 2,400 mm. (abt. 7′ 10″).
Height of stacked-up charge: 2,100 mm. (abt. 6′ 11″).
Outer diameter of coiled strips: 1,000-1,200 mm. (abt. 3′ 3″ to 3′ 10″).
Inner diameter of coiled strips: 350-500 mm. (abt. 14″ to 20″).
Power rating of the electric fan drive: 1.2 kW.
Fan speed: 500 r.p.m.
Computed consumption of electric energy per run: 215 kWhr/ton of charge.
Consumption of inert gas per ton of charge: 8.3 cub. m. (294 cu. ft.).
Water consumption per ton of charge: 0.5 cub. m. (17.7 cu. ft.).
The capacity of the furnace PSK 180 is rated at 180 kW.

The test data referred to below were obtained with the annealing of steel strip 350 mm. wide (abt. 13¾″) and 0.8-2.52 mm. thick (abt. 0.0315-0.1″). The weight per coil was 1300 kg (2860 lb.) with an outside diameter of 1,000 mm. (abt. 3′ 3″) and an inside diameter of 500 mm. (abt. 20″). The weight of the total charge was 7-8 metric tons. According to specification, the physical properties and the microstructure of the annealed product were to be: Tensile strength, 28-33 kg per sq. mm. (38,760-46,800 lb. per sq. in.), Elongation not less than 40 per cent, ASTM Grain size No. 5-7.

The properties of the steel strip after cold rolling and before annealing are as follows:

Thickness of strip		Tensile strength		Elongation
mm.	(inches)	kg/sq. mm.	(lb/sq. in.)	%
0.8—0.9	0.0315—0.0354	73.1—59.4	103,762 —84,348	6.1—3.2
2.52	0.1	64.7—63.0	91,874 —89,460	6.0—4.5

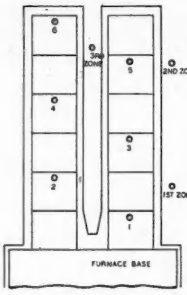


Fig. 2
Arrangement of thermocouples in strip coils.

The following points were investigated : Speed and time of heating of the strip in the furnace PSK-210 ; speed and time of cooling of the strip under the bell, and in air after removal of the bell ; accurate determination of the control temperatures required in the heating zones ; establishment of the difference between temperature indicated by the thermocouple in the base and the actual temperature of the strip ; change in temperature in the coils with the height and throughout the coils. The latter data were obtained by means of thermocouples specially provided at the points indicated in Fig. 2 ; the temperatures were registered with a potentiometer type recorder.

Seven test runs were made at various temperatures. In the first run, the furnace temperature was held at 800 deg. C. in all three furnace zones. This resulted in a temperature difference of 110 deg. C. between top and bottom of the charge. In the second, third, and fourth run, the temperature in the top zone was lowered. This decreased the temperature difference between top and bottom of the charge to 30-50 deg. C. In the fifth test, identical temperatures at top and bottom of the charge were achieved by increasing the temperature in the bottom heating zone to 850 deg. C., lowering it to 650 deg. C. in the second heating zone, and holding it at 720 deg. C. in the centre zone.

In the sixth test run, the same temperature distribution was employed, but after heating to 700 deg. C., the heating elements were switched off, while the bell was not lifted off the charge until two hours later. This procedure yielded a temperature difference in the charge of only 15 deg. C. The change in the average temperature of the charge throughout the full operating cycle is charted in Fig. 3.

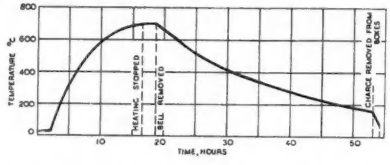


Fig. 3. Temperature variation in actual operating cycle of furnace of type PSK-210 with a charge of 7.5-8.5 metric tons.

With a charge of 7.0-8.5 tons of strip, the following operating times were required :

(1) Bringing charge in place and lowering bell, 1 hour.

(2) Blowing through and introducing inert gas, 1 hour.

(3) Heating, 14-16 hours.

(4) Bell remaining in place with heating current turned off, 2 hours.

(5) Cooling of charge with bell removed, 32-36 hours.

(6) Removing charge, 1 hour.

Total operating time, 51-57 hours.

With this mode of operation, the power consumption data are exemplified by the following figures :

Weight of charge, 7.6 tons.

Heating period, 15 hours.

Power consumption in first heating zone, 1368 kWhr.

Loading of first heating zone, 91.2 kW.

Power consumption in second heating zone, 108 kWhr.

Loading of second heating zone, 7.2 kW.

Power consumption in third heating zone, 300 kWhr.

Loading of third heating zone, 20 kW.

Total power consumption, 1776 kWhr.

Power consumption per ton of charge, 233.7 kWhr/ton.

DETERMINATION OF VERY SMALL OUT-OF-BALANCE FORCES.

(From *Elektrotechnischer Anzeiger*, Berlin, Vol. 60, No. 6, 17th March, 1943, pp. 63-64.)

BALANCING of small bodies, e.g., discs, is rather difficult, and cannot be accurately performed with the usual type of apparatus available to-day. For accurate measurements, it is essential that the masses rotating together with the sample should be light, and that only small forces should be absorbed by friction, braking or damping effect. If these losses exceed a certain limit, very small out of balance masses will not produce oscillations, hence their position and magnitude cannot be determined.

The apparatus described in the following is particularly suited for determining very small out-of-balance forces.

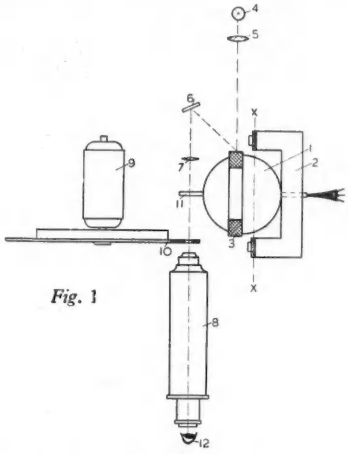


Fig. 1

The small body to be examined is in the case of Fig. 1 a disc (1) with a fixed spindle. The spindle is held by a light frame (2) which is spring supported in the x-x axis. To determine the position of the out-of-balance force, a light ring with a colour scale is mounted on to the disc (3). The colour scale (alternatively a graduated scale can be used) is illuminated from a source of light (4) through a lense (5). The light beam is reflected from the colour scale to a mirror (6), and is finally focussed by the microscope (8). A horizontal stroboscopic disc (10) is placed in the path of the light way ; (9) is a motor driving the disc (8).

The spindle (11) starts oscillating during the test. These oscillations can be observed through the microscope, and their amplitude can be read off a scale provided on the microscope lense. The operator determines the colour corresponding to the highest position of the spindle (11). The amplitude of oscillation is a measure of the out-of-balance force.

The operation with a graduated wing, in place of the colour scale, is very similar—it is only necessary to fit another lense (7) in the system in order to reverse the picture so that the figures on the scale can be read off.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- NEW EQUIPMENT
- BOOKS AND CATALOGUES
- BUSINESS CHANGES
- PERSONAL
- NOTES

BRITISH STANDARDS.

(Copies of British Standards may be obtained from the British Standards Institution, 28 Victoria Street, London, S.W.1).

NEW BRITISH STANDARD FOR TEST SIEVES.

THE British Standards Institution has recently issued a revision of the B.S. 410 for Test Sieves.

Particular attention has been paid in this revision to the question of the tolerances on the apertures, as a result of which modified tolerances on "average" and maximum apertures have been adopted which are more evenly graded throughout the series. Furthermore, an additional tolerance has been introduced called the "intermediate tolerance" and the maximum percentage of apertures that may exceed this intermediate dimension is specified.

It had been appreciated that even these revised tolerances might permit of too great a variation in sieving results to meet some special requirements such as may exist in research and control work, and the new specification includes a second series of sieves with more stringent tolerances designated "Special Test Sieves". This special series of sieves will not, however, be manufactured during the war.

The possibility of replacing the medium mesh wire cloth sieves by perforated plate sieves has received attention, but up to the present it has not been found possible to complete the research work necessary to establish the required data, and in the present revision wire cloth has been retained for the medium series. The manner of expressing tolerances on the apertures has, however, been brought into line with that now adopted for the fine mesh sieves.

Perforated plate sieves of smaller apertures than were formerly standardised have been included in the coarse sieve series and this series has also been extended to include a selected number of larger apertures.

As formerly, the revised specification provides for sieves of 8, 12 and 18 inches diameter, and in addition, particulars are now given of 3 inch diameter sieves as used for wet sieving.

An important feature of the revised specification is the comprehensive appendix relating to the method of examining the aperture widths and wire diameters of fine and medium mesh test sieves. This appendix, which has been prepared in consultation with the National Physical Laboratory, indicates the type of projection apparatus required and describes in detail the manner of carrying out the measurements and the number of apertures which have to be measured to determine whether sieves conform to the requirements of the specification.

In a further appendix to the specification particulars are given for general information of other sieve series in common use.

APPENDIX A TO B.S./STA. 7 ECONOMY OF SOFT SOLDERS.

THE above Appendix which has just been issued by the B.S.I. for the Ministry of Supply deals with specifications for solders, particularly from the aspect of economy in the tin content.

The recommendations have been prepared by an Advisory Panel of the British Non-Ferrous Metals Research Association under the Chairmanship of Dr. H. Moore. The solders are classified in four series.

Sub-Series A.—Lead-silver alloys. There are three solders in this series.

Sub-Series B.—Lead-tin-antimony alloys. There are two solders in this series.

Sub-Series C.—Lead-tin and lead-tin-antimony alloys with small additions of silver. There are six solders in this series.

Sub-Series D.—Lead-tin and lead-tin-antimony alloys. There are five solders in this series.

Guidance as to the choice of an appropriate solder, and suggestions for the uses for which the various solders are appropriate are given.

The Appendix is intended for insertion in the folder in which B.S./STA. 7. has been issued, and copies are available at a cost of 6d. each, or 3d. each for orders of twelve or more.

B.S. 1113-1943.—WATER TUBE BOILERS AND THEIR INTEGRAL SUPERHEATERS.

THE range of published British Standards for Boilers has been completed by the issue of a British Standard Specification for Water Tube Boilers and their Integral Superheaters. This Specification applies to water tube boiler units including superheaters, economisers, and to other parts connected to the boiler without the interposition of a shut-off valve which is exclusive of brickwork setting and insulation.

Details of materials specifications are included covering plates and rivet bars, seamless forged drums, forgings other than seamless drums, tubes and heaters, pipes and steel castings. Complete formulae are included governing the scantlings of the boiler and these formulae apply to boilers working under average normal conditions of draught, good feed water and adequate supervision.

Where working conditions are adverse, e.g., abnormal evaporation, bad feed water, exposure to the elements, or where of necessity maintenance supervision will be inadequate, it is recommended that the scantlings found by calculation from the formulae given should be increased.

The simplest method of compensation for adverse conditions of the kind is to design for a working pressure above that at which it is intended to operate the boiler, having regard to the probable reduction of life in service due to the adverse causes.

It has been the general desire to avoid the use of complicated formulae and this has been possible by confining consideration to the standard types of water tube boilers in general use. It is realised that some of the formulae adopted could not be applied to all conceivable combinations and variations in design which progress might dictate and in such cases the scantlings should be the subject of agreement between the purchaser and the manufacturer.

It is recognised that the design of water tube boilers is the subject of continuous development, and it is intended to review this Speci-

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fication annually with a view to the incorporation of such modifications as are found desirable as a result of progress. Price 7/6, post free.

HEATING AND HOT WATER SERVICE INSTALLATIONS.

THE above Code, which has been prepared under the Codes of Practice Committee set up under the aegis of the Ministry of Works, has just been issued for the Committee by the B.S.I.

This Code, which is a War Emergency Code, gives recommendations as to the most suitable forms of heating installations to be adopted at the present time and sets out the guiding principles to be considered in deciding on the type and size of system to be installed. The desirability of securing maximum economy not only in the labour and materials used, but also in the fuel and power required for maintenance has been kept prominently in view. Price 1/-.

P.D. 159 Amend. September 2nd, 1943, to 3.W.6 Flexible STEEL WIRE ROPES FOR KITE BALLOON CABLES.

THE British Standards Institution has just issued an addendum slip to the above Specification, which provides for an additional size of flexible cable which has a minimum breaking load of 1 ton. The diameter of the rope is approximately .155 inches and its weight 38 lb. per 1,000 feet.

P.D. 165. W.E. Amend. 1 : Sept., 1943, to British Standard Specification for Methods for Testing Glues. B.S. 647 : 1938.

The British Standards Institution has just issued an amendment to B.S. 647. Methods of Testing for Glues. In view of the shortage of walnut the amendment allows for the use of English beech veneer for the determination of joint strength in shear.

THE IRON AND STEEL INSTITUTE.

The next year's Annual General Meeting of The Iron and Steel Institute will be held in London on May 11th, 1944.

THE INSTITUTE OF WELDING.

Programme of Meetings to be held at the Institution of Civil Engineers, Great George Street, Westminster, S.W.1.

Wed., February 9th, 1944, at 4 p.m.
"Flame Cutting in the Shipyard." Paper by G. M. Boyd, A.M.I.Struct.E.

Wed., March 1st, 1944, at 5.30 p.m.
"Recent Developments in the Welding of Light Metals." Paper by W. K. B. Marshall, B.Eng.

Wed., April 12th, 1944, at 5.30 p.m.
"The Fabrication of Medium Sized Components." Paper by K. K. Doherty

Wed., May 17th, 1944, at 5.30 p.m.
"Factors controlling the Weldability of Steel." Paper by L. Reeve, Ph.D., B.Sc.

The British Institution of Radio Engineers.—Addressing the Institution on the 28th inst., Mr. J. L. Baird described the development of colour and stereoscopic television since he first demonstrated both colour and stereoscopic systems in 1928, when 3-aperture spirals on disc were used in conjunction with colour filters and colour discharge lamps. His present system and the 600 lines system requires the use of coloured glasses by the viewer. Mr. Baird described the numerous experiments he had conducted during the past 15 years and stated that his systems were now sufficiently advanced to be used for public service if facilities were available. He considered that within 5 years after the war these systems will be extended to the whole of Great Britain by means of radio links. For television transmission, he felt that for this new system no alteration in television receiving apparatus would be necessary and that only minor changes in the transmitters would be required. For the purpose of showing up-to-the-minute films of immediate public interest, Mr. Baird was of the opinion that when television broadcasting recommenced cinemas would show regular television "news reels." Further, 3-colour 500 lines system would, in Mr. Baird's opinion, be used in the television-telephone.

THE NATIONAL SMOKE ABATEMENT SOCIETY. AT the General Meeting and Conference of the Society held in London on November 5th, Mr. Will. Melland, formerly an Alderman of the Manchester City Council, was elected President in succession to the late Dr. H. A. Des Voeux.

Sir Lawrence Chubb, who is well known for his work with the Commons, was elected Hon. Treasurer of the Society.

At the Conference, which met to discuss "Smoke Prevention in Relation to Plans for Post-War Reconstruction," the following resolutions were adopted :

(1) THAT this Conference, recognising the very considerable advances in the prevention of air pollution that can readily be attained during reconstruction, calls upon the Government to give the problem the urgent and full attention that it warrants, for the purposes of materially improving public health and amenities, reducing a gross economic waste, and assisting the better utilisation of our fuel resources.

(2) THAT Local Authorities throughout the country be urged to appoint special smoke abatement committees for the purpose of ensuring that full local examination of the problem in relation to reconstruction, and recommendations for action, may be made, with particular regard to town-planning, the establishment of smokeless areas, the zoning and control of industrial establishments, and the smokelessness of new housing ; and that, further, Local Authorities be requested to support the efforts of the National Smoke Abatement Society and the Regional Smoke Abatement Committees in making appropriate representations to the Government.

CATALOGUES RECEIVED.

"Coborn" Electric Tippers.—Messrs. George Cohen Sons & Co. Ltd., have issued an Instruction Book on installation, operation, construction and maintenance of the "Coborn" and "Coborn Junior" electric tool tipper machines. The booklet (48 pages) contains general instructions for the workshop and also information on capacity, performance, dimensions and shipping specification to be used by Production, Design and Planning departments. Operations which can be performed with the electric tool tipper, include the following : Tipping of tool shanks with Carbide or Stellite. Tipping tool shanks with High Speed steel. Tipping of inserted tooth milling cutters with Carbide, Stellite or High Speed steel. Flash-butt welding of lathe tools.

Precision and Production Engineering.—Messrs. Alfred Partridge & Co. Ltd., Bredbury, Stockport, have published on the occasion of the 130th anniversary of the firm a technical catalogue, which is to perform a liaison service by giving a brief survey of the activities of the firm as precision engineers. Section 1 of the catalogue deals with high grade machined parts produced in quantity repetition work, and specialities ; Section 2 with the manufacture of jigs, tools and fixtures for modern production to technical laboratory standards. The beautifully illustrated catalogue is representative of the high standard of Messrs. Partridge.

Oil Seals and Oil Retainers. A well illustrated 12-page catalogue recently issued by Messrs. Charles Weston & Co. Ltd., Torrington Avenue, Coventry, gives details of Gits oil seals and Weston oil retainers. The catalogue includes a Table of standard sizes and installation instructions.

Rapidor Machine Tools and Accessories. An abridged catalogue of 16 pages, published by Messrs. Edward G. Herbert Ltd., Atlas Works, Levenshulme, Manchester, 19, presents general information on "Rapidor" machine tools and accessories. 29 types of drilling, grinding, milling, screwing, tapping, sawing, and other metal working machines and their accessories are described and illustrated.

PERSONAL.

Mr. A. C. Allen and **Mr. N. C. Robertson** have been appointed Directors, and Mr. A. W. Martin Chief Engineer of Messrs. E. K. Cole Ltd., Aston Clinton, Bucks.

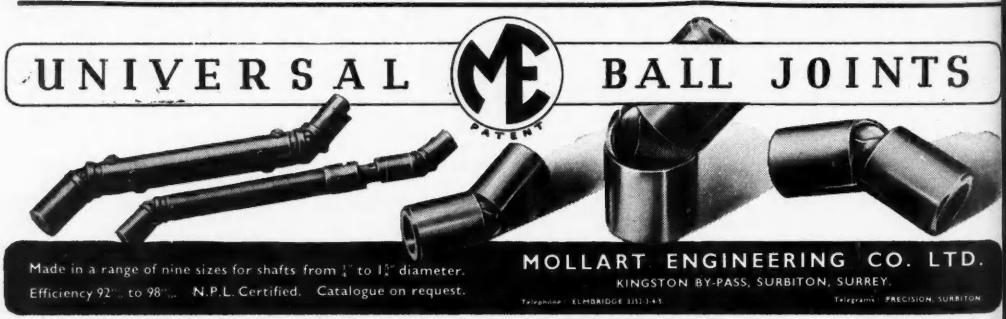
Mr. Norman Burke and **Mr. C. E. H. Eckersley** have been appointed directors of James Howden & Co. Ltd.

Captain G. F. Davies, R.A., has been appointed a director of Messrs. Hick, Hargreaves & Co., Ltd.

Mr. A. S. Gilley, A.M.I.Mech.E., has been appointed General Manager to Messrs. Fawcett, Preston & Co., Ltd.

Mr. W. Glass, A.M.I.Mech.E., has been appointed Deputy Managing Director of Messrs. Johnson and Philipps, Ltd. **Mr. S. J. Passmore** and **Lt.-Col. R. W. C. Reeves, A.M.I.E.E.,** have been elected directors of the Company.

Mr. J. J. A. Jones, Manager of the Central Research Department of the United Steel Companies, died on November 1st at the age of 49.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